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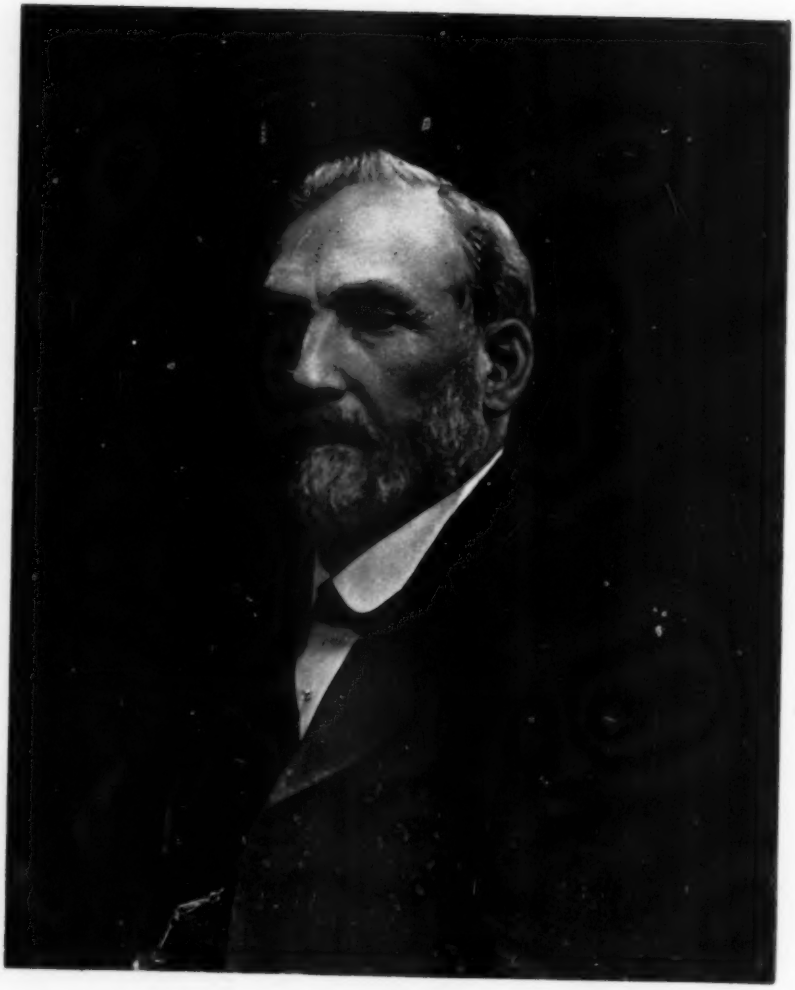

Edwin Reynolds

PAST PRESIDENT
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HISTORY OF THE AMERICAN SOCIETY OF MECHANICAL ENGINEERS

PRELIMINARY REPORT OF THE COMMITTEE ON SOCIETY HISTORY

CHAPTER X

CONTINUED GROWTH OF THE SOCIETY

244 Following upon the World's Fair of 1893, and perhaps largely because of it, the Society experienced a period of great activity and progress. The business depression gradually passed away, and with the commercial revival the membership of the Society increased, and the extent to which its members were engaged in the development of the country is reflected in the Society's activities as represented in its Transactions.

245 The reëlection of Eckley B. Coxe as President for the year 1894 retained in the chair a man already experienced in the work of the Society, while the general interest of the leading members gave added impetus to the efforts of the administration.

246 Following the custom, now become general, of basing his presidential address upon the department of engineering work with which he had been most closely identified, Mr. Coxe presented what proved to be a most valuable discussion of the possibilities of small sizes of anthracite coal for the generation of steam. A special interest attaches to this address at the present time (1909) because it has been assumed in many quarters that the realization of waste in the use of the great natural resource of coal is a very recent idea, whereas, prior to Mr. Coxe's work in this direction, which began in 1890, the waste of anthracite coal was very fully appreciated. Mr. Coxe himself

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The Preliminary Report will appear in The Journal of the Society from month to month, and thus enable the matter to be open to comment during its completion. It is especially desired that any member who may be in the possession of facts or information bearing upon the various points as they are thus made public will communicate with the committee, in order that the final and completed report may have the advantage of the collaboration of the membership at large.

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served upon a commission which reported to the State of Pennsylvania in 1893, and the situation as it was then presented convinced him that what has later been called "conservation" was a matter in which he as a coal operator was deeply interested. A quotation from Mr. Coxe's address will show very briefly the situation as it was presented to him:

247 "When I first remember the anthracite coal business, all the sizes below stove coal were considered of little or no value. Chestnut coal was used to a greater or less extent about the collieries for making steam, but much of it was thrown away. It gradually became a domestic fuel, and pea coal took its place for generating steam. This size is gradually coming into use (1893) for domestic purposes, and buckwheat coal is taking its place for steam. Two smaller sizes, known as rice and barley, are already being used to a large extent for the same purpose. The value, or cost, of these coals diminishes very rapidly with the size."

248 It was to increasing the value, considered as a steam fuel, without materially increasing the cost of these small sizes, that Mr. Coxe devoted much time and effort, and his address may well be examined by those who have been led to believe that the engineer needed to be encouraged to minimize waste in the use of coal.

249 Increased efficiency in the generation and use of power as well as in the conduct of machine-shop operations formed topics toward which many members of the Society were directing their attention, and the Transactions of 1894 contain some important papers on these subjects. Thus, at the New York Meeting of 1893, papers on steam economy predominated, these including contributions by Professors Aldrich, Carpenter and Jacobus on the steam indicator, by Professor Carpenter on water-hammer in steam pipes, and by Mr. Pike on the influence of steam piping on the efficiency of steam plants. At this meeting Professor Thurston presented an important paper reviewing the contemporary economy of the steam engine, based upon the high performance of the Milwaukee pumping engine, and showing that a thermal efficiency of more than 19 per cent had been attained. Another paper of importance and of far-reaching influence presented at this meeting was that of F. W. Taylor, containing notes upon the actual behavior of belting in the machine shop, this paper having a direct bearing upon the methods afterwards so extensively undertaken to increase the output of machine tools by the introduction of the heavy cuts and feeds possible with high-speed steels.

250 The close of the year 1893 was marked by the first break in the ranks of the distinguished men who had served as president of the Society, George H. Babcock passing away on December 16. A man of broad professional culture, with many activities in the field of mechanical engineering, he will doubtless be best remembered for his part in the development of the water-tube boiler devised by himself and his life-long associate, Stephen Wilcox.

251 The summer meeting of the year 1894, held from June 5 to 9, was the first convention of the Society to be held beyond the limits of the United States, the meetings occurring in the city of Montreal, Canada. Here the Society received a hearty welcome from the official representatives of the city government, the Canadian Society of Civil Engineers, and the governors and faculty of the McGill University. Among the entertainments especially noteworthy at this meeting may be mentioned the reception at the residence of the Chancellor of McGill University, Sir Donald A. Smith, afterwards Lord Strathcona; and the garden party held at the Government House, Ottawa, by Lord Aberdeen, Governor General of the Province.

252 The second address of Eckley B. Coxe, delivered in the hall of McGill University, was very appropriately devoted to the subject of technical education, emphasizing the importance of the combination of technical and business training as essential to success in the engineering profession. The trend of the papers at the Montreal meeting again showed the attention which was being given to steam engineering. The discussions included papers on the steam jacket, the influence of compression on engine performance, the cylinder proportions of compound engines, and steam distribution. Papers on the cost of power, engine tests and engine details, showed the extent to which the principal prime mover occupied the thoughts of the members, and the activity of the discussions testified to the interest which the papers created.

253 The annual meeting of 1894, held as usual in the house of the Society in New York, contained, in its business discussions, some matters of much interest. During the winter of 1894, a series of monthly meetings had been held for the discussion of technical subjects, these meetings having taken place at the house of the Society, and the expenses being defrayed by subscription of the members interested.

254 An active discussion took place at this annual meeting as to the advisability of making such meetings a part of the regular work of the Society, the principal feature of the discussion being the desirability of avoiding the creation of an impression that the New York

members were receiving special advantages over distant members of the Society.

255 The result of the discussion was the adoption of a plan for the continuance of the meetings, but after a time these monthly gatherings were abandoned without any special action of the Society to that effect being taken.

256 A report of the committee on standard gauges for the thickness of wire and metal, presented a progress report advocating the use of a system based upon the actual thicknesses in the decimal system, as opposed to the older arbitrary numbering gage systems.

257 It is interesting to note that at this meeting it was reported that of the $32,000 issue of bonds disposed of by the Society in connection with the purchase of the house at No. 12 West Thirty-first St., New York, $19,000 worth had been acquired by the Society, by purchase, by surrender in exchange for life memberships, and by gift and bequest.

258 It has been noted, in commenting upon the papers presented at the various meetings, that certain departments of engineering work appeared to predominate at certain periods of time. At the annual meeting of 1894 the principal technical subject seemed to be that of the materials of construction, both as regards the resistances to various kinds of stresses, and as to the nature of those stresses. Among the more important of these papers may be mentioned the monograph of W. J. Keep upon tests of cast iron, this having been prepared for the Committee on Uniform Methods of Testing Materials, and being the first of the notable contributions of Mr. Keep to this subject. Other papers relating to the strength of materials include those of Professor Lanza, tabulating the results of tests made at the Massachusetts Institute of Technology upon full-sized spruce columns, by L. S. Randolph on the strength of railway car axles, and by G. M. Sinclair on steel forgings. At this meeting M. P. Wood presented the second of his extensive papers on rustless coatings for iron and steel, continuing the discussion commenced at the Montreal meeting. Other related papers treated of the proportions of springs, of rail pressure of locomotive driving wheels, and the stresses in the rims of pulleys and flywheels. At this meeting the election of E. F. C. Davis as President of the Society was announced.

THE HIGH-PRESSURE FIRE-SERVICE PUMPS OF MANHATTAN BOROUGH, CITY OF NEW YORK

DESCRIPTION OF PUMPS AND PUMPING SYSTEM WITH RESULTS OF TESTS

By Prof. R. C. Carpenter, Ithaca, N. Y.
Member of the Society

The object of this paper is to present a concise description of the high-pressure pumping system installed for fire service in the city of New York and the results of a test of the pumping machinery.

2 The system protects the district extending north from City Hall to Twenty-fifth Street, and east, approximately, from the North River to Second Avenue. It comprises about 55 miles of extra heavy cast-iron main, from 12-in. to 24-in. in diameter, with 8-in. hydrant branches; and two pumping stations so located that they never can be in the center of a conflagration. At the present time the pumping stations have a combined capacity of over 30,000 gal. per min. delivered at a pressure exceeding 300 lb. per sq. in.

THE SOURCE OF WATER SUPPLY

3 The supply of water is ordinarily obtained from the water mains of the city, which deliver Croton water to the stations at a pressure of from 14 lb. to 40 lb. per sq. in., depending upon the demand for water in that district. Both of the pumping stations are located close to tidal water and connections are made so that sea water can be obtained in case of difficulty with the Croton supply.

4 The advantage of the Croton water over salt water is that it is less likely to injure goods, and as the amount required for fire purposes is only a small percentage of that consumed for the daily supply of the city its use for fire protection makes no material difference from financial or insurance standpoints. As this is a matter of considerable importance data upon the quantity needed are given in the next paragraph.

To be presented at a meeting of The American Society of Mechanical Engineers. All papers are subject to revision.

WATER REQUIRED FOR FIRE PURPOSES

5 The general impression that an enormous quantity of water is required for fire purposes is erroneous as shown by figures furnished to chief engineer I. M. de Varona by the fire department for the Boroughs of Manhattan and Brooklyn, years 1900, 1901, 1902, 1903 and 1904. These give the average quantity of water used for fire protection during these years in the Borough of Manhattan as 74,010,803 gal. per year, of which 31,056,928 gal. was river water. The daily average use of Croton water, therefore, for the above five years was 117,000 gal.

6 For the Borough of Brooklyn the average for five years was 43,705,568 gal. of which 19,010,928 gal. was river water; daily average, 67,000 gal.

7 During these five years the greatest quantity used in the Borough of Manhattan was 99,000,000 gal. in 1901, which included 69,500,000 gal. of river water, leaving 29,500,000 gal. for fresh water, and Mr. de Varona states (Report of the Department of Water Supply, Gas and Electricity): "Even if this quantity be made 100,000,000 gal. per year, by comparing it with the average daily consumption of about 300,000,000 gal. it will be seen that the total amount used for fire purposes would be only about one-third of the amount used for all purposes in 24 hr., forming, therefore, an insignificant percentage of the total consumption. The quantity needed for fire purposes (one-tenth of one per cent) may therefore be entirely neglected as a factor in determining the water supply of the city.

8 "The capacity of each of the pumping stations will be for the present 15,000 gal. per min. or 43,000 gal. per day for the two stations. By the installation of three additional units in each station, for which provision is made, this capacity can be increased in round numbers to 69,000,000 gal. per day.

9 "The two stations, with the motors and pumps as installed, have a total capacity in excess of that of all the fire engines in the Boroughs of Manhattan, the Bronx and Brooklyn working under normal conditions. This comparison assumes the engines to work on one line of 2½-in. hose, say 500 ft. long, under a pressure of, say 200 lb., and with the capacities as printed in the official blank forms of the reports of the fire department. It should furthermore be remembered that provision is made for the installation of still another pumping station."

MOTIVE POWER

10 The power for driving the pumps is transmitted electrically from several of the electric power and lighting systems located on Manhattan Island. As the stations of systems are widely separated and any or all of them are available for motive power the system of electric transmission was considered more reliable in the case of a large and general conflagration than power plants maintained directly at the pumping stations. Each station is provided with two independent sets of transmission lines located as far as possible beyond danger or injury in case of a great conflagration.

11 The cost of erecting and maintaining an independent power plant would have entailed a greater annual charge than the cost of the electric current; consequently the present arrangement is advantageous from a financial standpoint.

12 In addition to the charge per kilowatt for the current delivered there is a charge aggregating $90,000 per year for reserving the first right of use for the necessary generating machinery for this purpose. The total cost of maintenance of the system is estimated at $170,000 a year, which amount it is believed will be saved many times over by a reduction in insurance premiums now paid in the protected district.

13 The electric current is supplied at a pressure of 6600 volts from the following stations of the New York Edison Company, having the capacity indicated: 53 Duane Street, 7600 kw.; 115 East 12th Street, 1700 kw.; 45 West 26th Street, 400 kw.; 140th Street and Ryder Ave., 4000 kw.; Waterside Station No. 1 and No. 2, 196,700 kw. In addition there are feeders extending to the Brooklyn Edison Company stations which can be called on in case of an emergency demand.

14 The pumping stations are connected to 18 sub-stations, equipped with rotary converters and storage batteries, aggregating a capacity of 124,000 ampere hours at 135 volts, an enormous reserve.

15 Each station is connected with the main stations of the Edison Company by two 250,000 c.m. three-phase cables laid in ducts, and two independent reserve feeders extend to the sub-station system of the Edison Company. With all these precautions, interruption of the power supply would seem a physical impossibility.

THE DISTRIBUTION SYSTEM

16 The following information upon the distribution system is taken largely from the department report of Chief Engineer de Varona

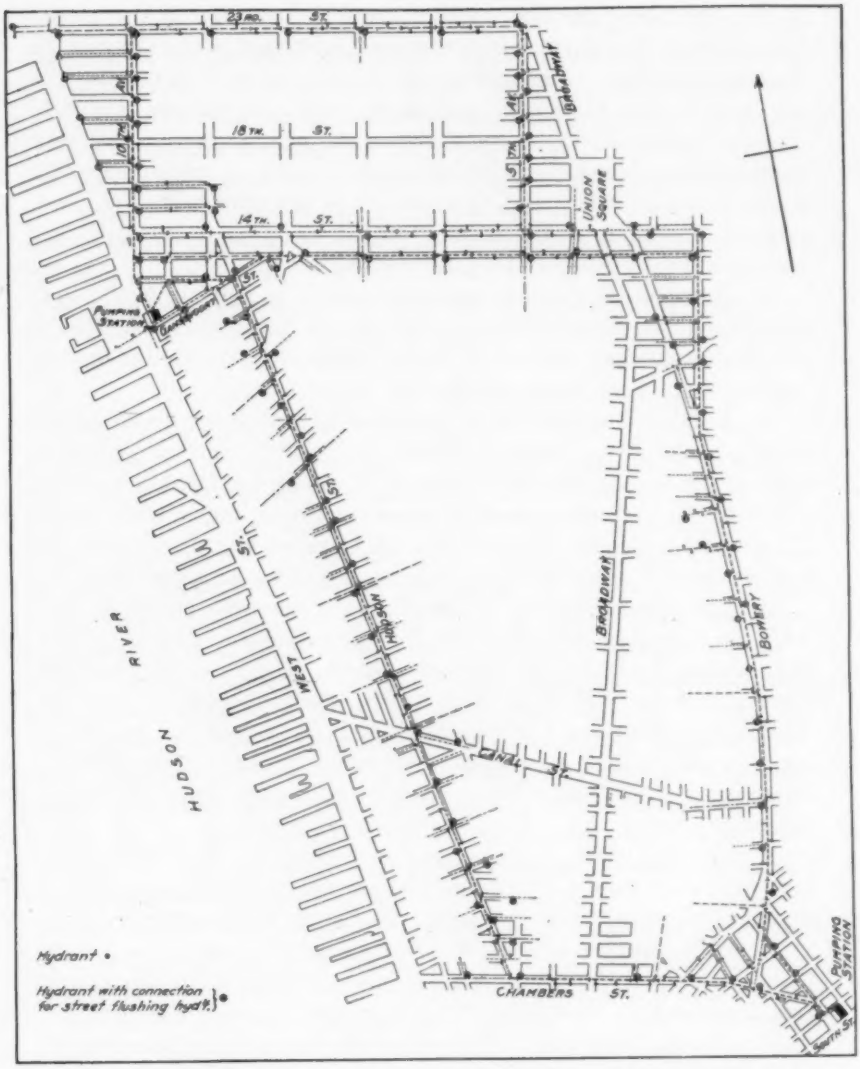

FIG. 1 SHOWING LOCATION OF STATIONS AND AREAS COVERED BY HIGH-PRESSURE PUMPING SYSTEM

THE AREA INDICATED IS SERVED BY A SYSTEM OF MAINS RANGING FROM 24 IN. TO 12 IN. IN DIAMETER WITH 8-IN. HYDRANT CONNECTIONS

for 1905. Fig. 1 shows the system to be bounded by mains laid on the north through Twenty-third Street; on the east, through Broadway to Fourteenth Street, through Fourteenth Street to Third Avenue, down Third Avenue to the Bowery, down the Bowery to Chambers Street; through Chambers Street on the south to West Street; and on the west through West Street.

17 The area actually protected is considerably greater than this as hose can be extended over a zone 600 ft. wide beyond the limits of the mains.

18 This district was selected as that in which the fire losses were the greatest and which most urgently needed fire protection. Plans have been prepared for the extension of the system southerly to the Battery, easterly as far as the East River, and, if necessary, northerly as far as Fifty-ninth Street, by the simple extension of the mains and probably the erection of a third pumping station.

19 The pipes, castings and hydrants were tested at a pressure of 450 lb. The specified allowance for leakage in a 10-min. test was at the rate of 4 gal. in 24 hr. for each lineal foot of pipe joint, equivalent to a leakage of 487,000 gal. for the whole system in 24 hr., which is somewhat over one per cent of the total specified pumping capacity now installed. The actual leakage on test was at the rate of 264,000 gal. per day or about six-tenths of one per cent of the pumping capacity. Considering the difficulties of construction and the high pressure the results attained were remarkable and reflect great credit on the engineer in charge.

20 There are sufficient hydrants so that if a block were on fire 60 streams of 500 gal. per min. each, or the full capacity of both stations, could be concentrated on a block with a length of hose not exceeding 400 ft. to 500 ft., assuming the use of 3-in. hose and 1¼-in. nozzles.

21 The layout of the mains at the stations both for suction and delivery is on the loop system; that is, the supply can be taken from either one of two mains, and the discharge is through either one or both of two mains. With this system even the breakdown of one of the discharge mains at the station would only slightly reduce the pressure at the fire and would not affect the capacity of the station, as the pumps would be capable of forcing their full discharge through the short length of a single 24-in. main made necessary by such an accident.

22 The mains are of cast-iron, bell and spigot pipe, of the thicknesses given in the following table:

Size of Pipe Inches	Thickness Inches	Unit Tensile Strain with 300 lb. pressure	Factor of Safety
24	1⅞	1920	0.4
20	1½	2000	10.0
16	1¼	1900	10.4
12	1	1800	11.1
8*	⅞	1371	14.6

* Used only for hydrant branches.

SUPPLY PIPING

23 At the South Street Station the fresh water supply is derived from two 30-in. lines, one connected at Chestnut Street to the 36-in. line on Madison Street, and the other connected at Pike Street to the 36-in. line on Division Street. These two main feeders, to which the two 30-in. lines are connected, increase to 48 in. in diameter and extend independently and directly to the Central Park Reservoir and are also reinforced by connections with the main feeders in this section of the city.

24 An auxiliary salt-water supply, consisting of two 36-in. pipes about 140 ft. long, brings the salt water from the East River to a suction chamber located directly in front of the pumping station. This suction is so constructed that the pipes are always below mean low water, thus insuring a supply at all times and avoiding the possibility of a break in the suction caused by air getting into the suction lines. On the river end of this suction there are constructed heavy bulkhead screens and in the suction chamber are two sets of bronze screens which are readily accessible for cleaning. From the suction chamber there are taken two 30-in. flanged mains to the duplicate set of mains in the pumping station proper. The vacuum in these 30-in. pipes is maintained by automatic electric vacuum pumps located on the pump room floor of the station.

25 At the Gansevoort Street Station the fresh-water supply is derived from two 30-in. mains, one connected at Twelfth Street to the 48-in. line on Fifth Avenue, which runs direct to Central Park Reservoir, and the other connected to the 36-in. line on Ninth Avenue at Little West 12th Street, which increases to a 48-in. line and runs also direct to the Central Park Reservoir. These two main feeders, in addition to having their supplies direct from Central Park Reservoir, are also reinforced by connections with the main feeders in this section of the city.

26 The salt-water suction lines for this station are practically identical with those for the South Street Station except that the 36-in. lines from the North River to the station are 650 ft. long.

PUMPING STATIONS

27 The two stations, known as the Gansevoort pumping station, located near Gansevoort Market on the North River, and the South Street station, located on the corner of Oliver and South Streets near the East River, are identical in construction and equipment. The buildings are of simple design, of steel fire-proof construction, with concrete foundations. The Gansevoort Street building, which is typical of both, is one story high with basement, 63 ft. 8 in. by 97 ft. 4 in. Each station is large enough for eight pumping units.

MACHINERY

28 There are now five units in each station consisting of Allis-Chalmers five-stage centrifugal pumps driven by Allis-Chalmers induction motors and the necessary auxiliary machinery. The motors and pumps are alike and their parts are interchangeable.

29 The pumps each have a specified capacity of 3000 gal. per min. of sea-water, working with a suction lift of 20 ft. and a delivery pressure of 300 lb. per sq. in. The actual capacity as indicated by a 24-hr. test was about 30 per cent in excess of that specified. The original specifications contemplated the use of six-stage pumps, with the expectation that sea-water would be used at each fire. Because of the facts already referred to (Par. 4), that the relative amount of water required for fire purposes is insignificant and that sea-water may do considerably more damage to goods than fresh water, a change in the specifications was agreed to whereby the pumps should work at best efficiency when receiving water from the Croton mains at a pressure on the intake side varying from 15 lb. to 40 lb. per sq. in.

30 To meet this new condition the pumps were all built with five stages. All the sea connections and priming machinery as originally contemplated were installed, so that sea-water can be pumped into the mains whenever desired. The effect of the change is merely to reduce the pressure head slightly in case sea-water is used.

ARRANGEMENT OF MACHINERY

31 The floor-plans of the buildings and general layout of machinery, piping, switchboards, etc., are shown in Fig 2. As will be seen

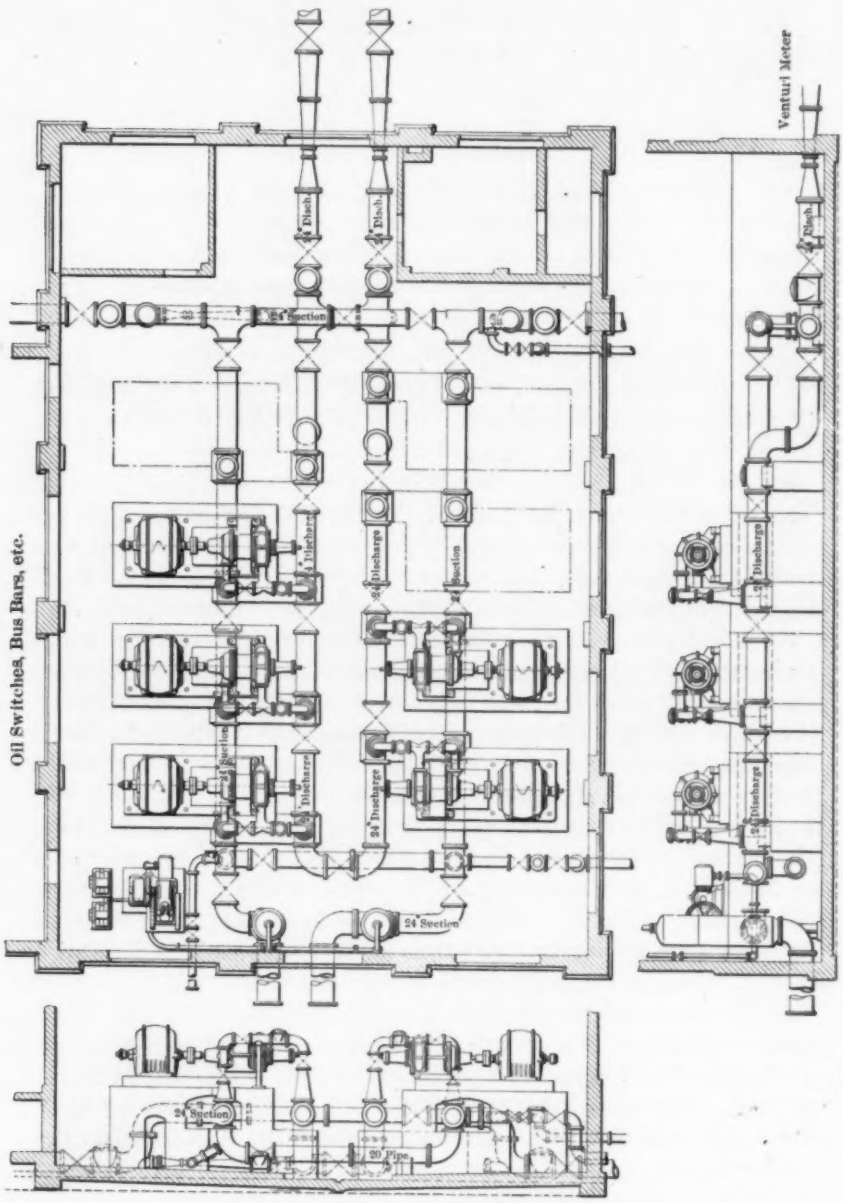

FIG. 2 PLAN AND ELEVATIONS SHOWING ARRANGEMENT OF HYDRAULIC AND ELECTRICAL APPARATUS IN PUMPING STATIONS

space is provided for three additional units. Working detail plans of the machinery were furnished by the contractor. The arrangement shown in Fig. 2 is the same for both stations, the only difference being that the switchboard and office in the South Street station are on different sides of the building as compared with the Gansevoort station.

32 The motors and pumps, with suction and delivery branches, are located on the main floor of the pump room. The switchboard and switchboard apparatus are placed in an enclosed two-story and basement gallery.

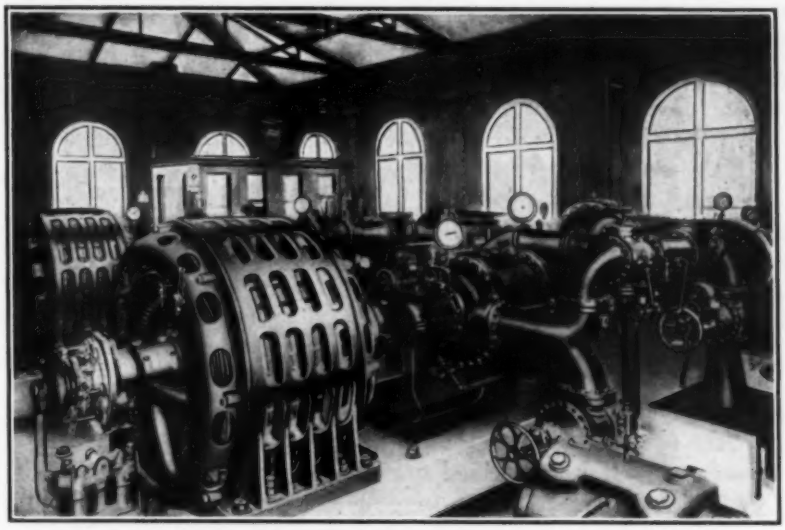

FIG. 3 INTERIOR VIEW OF STATION

33 The four high-tension feeders and all other wires entering the building are brought in through the gallery basement. All terminal work on the entering wires is located in the basement. On the first floor of the gallery, which is approximately on the same level as the pump-room floor, are placed the oil switches, with their controlling and protective devices, fire-proof cells and compartments.

34 The operating switchboard is conveniently located in the enclosing wall of the gallery, and is so placed as to allow a man standing on the pump-room floor to perform all the operations necessary for controlling the apparatus in the station. The bus bars,

with their fireproof compartments, are placed on the second floor of the gallery.

MOTORS FOR CENTRIFUGAL PUMPS

35 Each pump is direct-connected to its motor by a flexible coupling which takes care of any variation from alignment. The motors are of the constant-speed induction type, 3-phase, 25-cycle, 6300-volt to 6600-volt, designed to operate at about 740 r.p.m. The motors are of the wound rotor type, and in starting an iron grid resistance is connected in the secondary circuit and gradually cut out by means of a handwheel on the motor switchboard panel. When the resistance is all cut out the rotor is automatically short circuited and operated by specially constructed solenoids which are operated by a small switch mounted directly on the shaft of the handwheel above referred to. An interlocking arrangement prevents the operator from closing the switch connecting the motor to the line while the motor is short-circuited.

36 The specifications required the motors to have sufficient starting torque to attain full speed between 30 sec. and 45 sec. after starting, with a current not exceeding 150 per cent of that used when the motor is working under full speed. Each motor was required to develop not less than 800 b.h.p. when using current of 6300 volts, 25 cycles, and under these conditions to have an efficiency not less than 92 per cent, a power factor not less than 93 per cent, and a motor slip not in excess of 2 per cent. At three-quarters load the efficiency was not to be less than 92 per cent and the slip not to exceed 1.5 per cent. It was specified that the temperature of the motors should not rise more than 40 per cent on a 24-hr. test at full load, when measured by a thermometer, the air in the room being 25 deg. cent.

37 Prof. Geo. F. Sever of Columbia University tested two of the motors in the shops of the contractor and found them to meet the specifications and to have a full-load efficiency of 93.2 per cent. The other motors were inspected and found to be alike and were assumed to have the same efficiency. The motors were also tested for temperature rise at the time of the official test to be described later.

MOTORS FOR AUXILIARIES

38 Direct-current motors of 240 volts are provided to operate the various gate valves in the station and the piston pumps employed for maintaining a vacuum on the salt-water suction lines.

PUMPS

39 As previously stated the pumps were finally constructed with five stages, each to give a pressure of somewhat over 60 lb. per sq. in., making the combined pressure of the five stages about 300 lb. per sq. in. above the intake pressure, which is the maximum working pressure of the stations at normal speed of 740 r.p.m. This type of pump is the simplest now on the market for pumping water either against a high head or low head, and this simplicity was the deciding factor which led to the selection of this style of machinery.

40 The pumps are water-balanced by a piston connected to the last impeller and upon which the water pressure acts, but should any additional end-thrust occur, it would be taken up by the ball bearing provided in the outboard bearing. This ball bearing consists of two rings of 1½-in. diameter steel balls and is water-cooled. The balancing piston is fitted very loosely in order to keep the friction losses small, and as a result a considerable amount of water leaks past it into a chamber at the end of the pump, which is provided with a discharge pipe and valve leading into the suction. By adjusting the valve in this pipe the difference of pressures on the piston can be regulated as desired. The bearings are of the ring-oiled type and are separated from the pump casing by packing glands which prevent foreign matter from entering the bearings. The impellers are of bronze and the shaft of forged steel. All parts of the runners and diffusion vanes are thoroughly lubricated by oil cups on the base of the pumps, shown in Fig. 4. A feature is the wide base which allows the pump barrel to set low, giving stability.

41 Each combined unit is equipped with automatic and hand control. The pumps are kept primed for instant service and the simple operation of a switch on the main switchboard starts the machine and gives full pressure in about 30 sec.

PRESSURE REGULATING VALVES

42 A combined regulating and relief valve is interposed between the discharge pipe and the suction pipe of each pump, and set to regulate the discharge of each pump to any predetermined pressure. When the volume of the water discharged by the pump is in excess of that forced into the system, this valve acts as a relief valve and by-passes this excess into the suction to the pump, the pressure on the main distribution system remaining at the predetermined point.

When no water is forced into the distribution system all of the water discharged from the pump is then by-passed into the suction.

43 The pressure-regulating valves were made by the Ross Valve Mfg. Co., of Troy, N. Y., and much of the practical success of the station has been due to the accuracy with which they maintain any desired pressure. The construction of these valves at this particular time cannot, however, be given with fairness to the manufacturers.

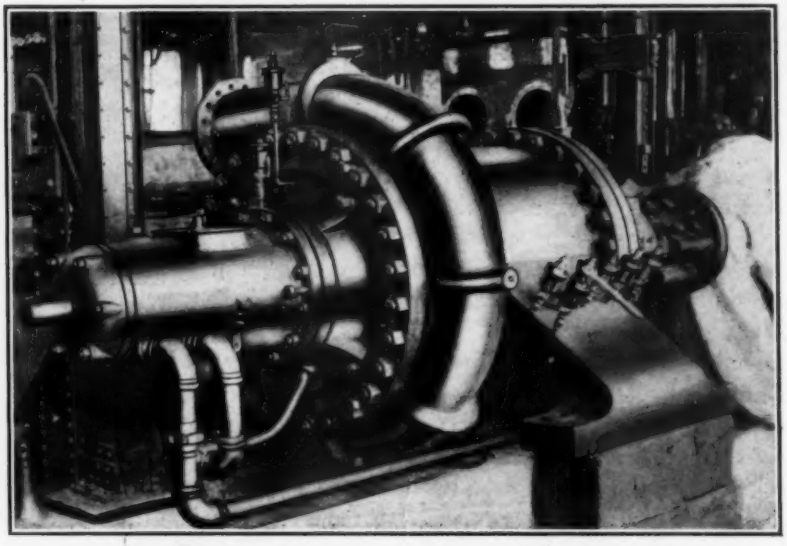

FIG. 4 MULTISTAGE PUMP, CAPACITY 3000 GAL. PER MIN.; MAXIMUM HEAD, 300 LB. PER SQ. IN.

PRIMING APPARATUS FOR SALT-WATER SUCTION LINES

45 The priming apparatus in each station consists of three motor-driven vacuum pumps, each arranged to maintain automatically a vacuum of 26 in. in the suction lines. These pumps are of the piston single-action type, one having a displacement capacity of 300 cu. ft. per min. for a piston speed of 200 ft. per min. and each of the others a displacement capacity of 50 cu. ft. with a piston speed of 160 ft. per min.

46 An air-collecting chamber is connected to each of the salt-water suction lines and equipped with water-gage glass and vacuum gage. The air-suction piping between the air chambers and the air pumps is provided with a vertical loop sufficiently high to prevent

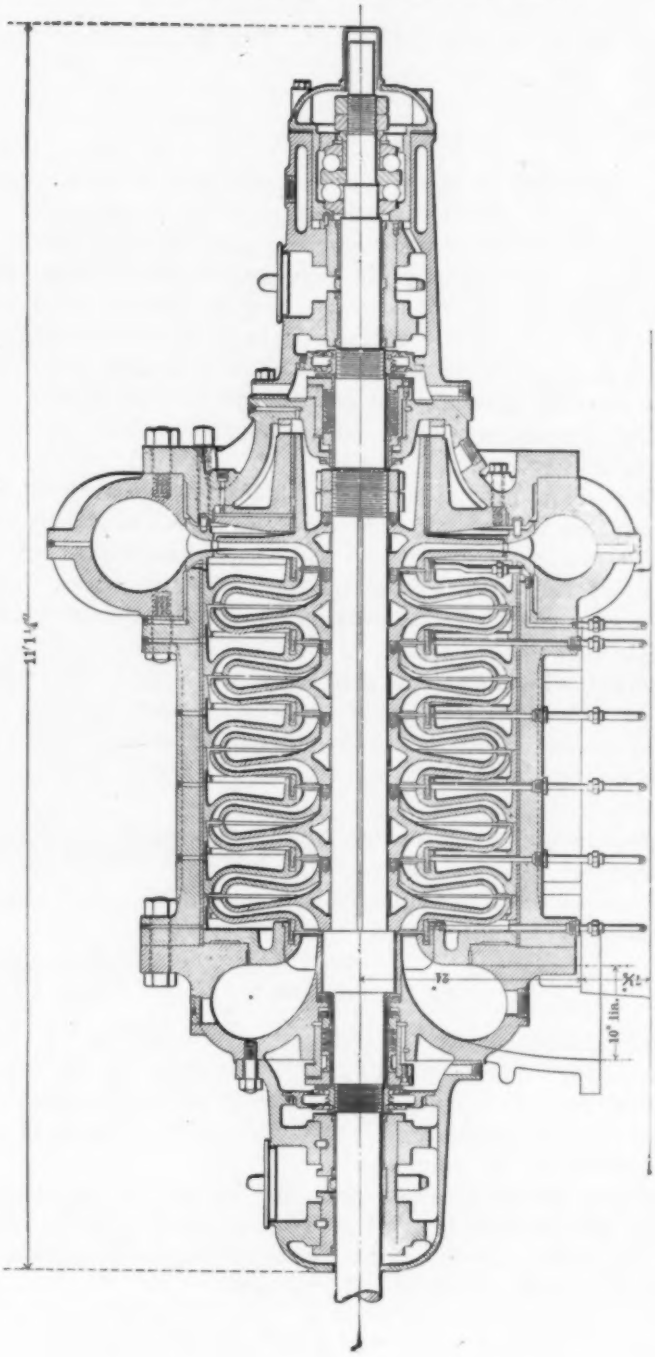

FIG. 5 CROSS-SECTION OF FIVE-STAGE, HIGH-LIFT CENTRIFUGAL PUMP

water being carried over to the pumps. The air pumps are interconnected to each air chamber.

VENTURI METERS

47 Venturi meters for measuring the discharge of water from the station and from one main to the other were set by the contractor on each discharge main and on the cross-connecting main. The meters of the discharge main are 24 in. in diameter and on the cross-over main 12 in. in diameter. These meters were standarized under the direction of F. N. Connet, chief engineer of the Builders Iron Foundry, Providence, R. I., and were provided with dial-indicating gages and also chart-recorders graduated to indicate the flow in gallons per minute; and in addition with an integrating meter which registers the total flow in gallons.

48 The readings during the test were taken by a mercury manometer, graduated to show the capacity in thousands of gallons per minute. For this purpose a Venturi manometer was attached with a temporary connection to each of the 24-in. Venturi tubes. The manometer gave essentially the same reading as the indicating dial on the main register.

49 The Venturi manometer is practically a tube partly filled with mercury, one side of which communicates with the upstream pressure chamber of the meter tube, while the other communicates with the throat-pressure chamber. The connections with the manometer are indicated in the diagram, Fig. 6.

50 The sketch shows a 24-in. high-pressure meter tube, its register-indicator-recorder and manometer. The instruments and meter tube are drawn to scale, but in the pumping station the meter tube is about 75 ft. distant from the instruments.

TESTS OF MACHINERY

51 The specifications for the pumping system provided for an endurance test of each motor and pump lasting 24 hr. without stop, during which time the capacity and efficiency of the pumps and motors was to be determined. The tests were to be in charge of an expert appointed by the commission.

52 The specifications provided for making the test with sea water, but this was later changed to a test with Croton water under the conditions of actual use. In view of this change the contractor increased the efficiency guarantee from 70 to 71 per cent.

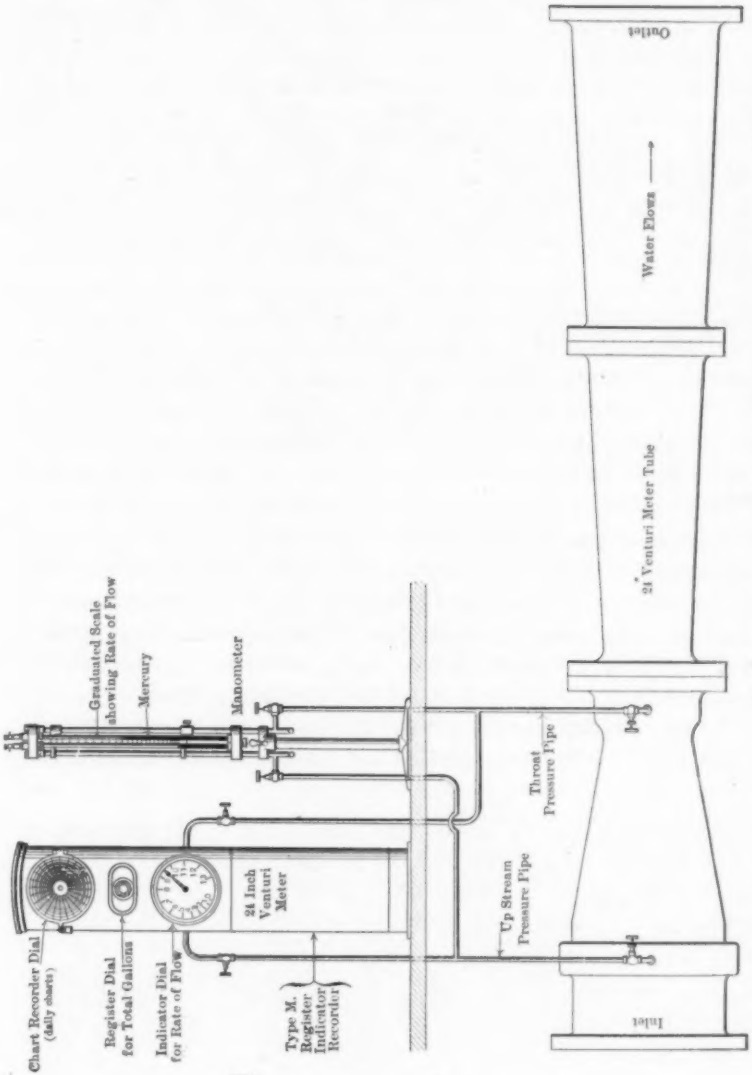

FIG. 6 ARRANGEMENT OF MANOMETER AND ITS CONNECTIONS TO THE VENTURI METER

53 The original specifications called for a capacity of 3000 gal. of sea water per minute against a discharge pressure of 300 lb. per sq. in. and a suction lift not exceeding 20 ft. The total increment of pressure is equivalent to 308.66 lb. from the intake to the delivery side. The Croton pressure varies at the stations in different parts of the day from about 40 lb. to 13 lb. per sq. in. and is affected by the amount of water being drawn from the mains. Consequently, to meet the requirements, the delivery pressure would need to be 308.66 lb. in excess of the intake pressure. There is also a further correction from the fact that sea water is heavier than fresh water and this correction under maximum conditions might amount to 2.5 per cent.

54 The specifications further provided that the brake horse-power developed by the motors under test should be computed from the electrical energy supplied to them, corrected for the efficiency of the motors as determined by the test. They further provided that if the aggregate of all stops exceeded one hour for any motor the test for capacity for such motor was to be run over again for a period of 24 hr.

55 The specifications also provided that the pumping capacity of the apparatus and the efficiency of the pumps should be based on the minimum rate of pumping during any eight consecutive hours of the endurance test, during which none of the motors were stopped.

56 The discharge of the pumps was determined by the reading of the Venturi meters, one of which was located in each discharge line. These readings were under the direction of F. N. Connet, and were checked by observers representing the contractors and also the city.

57 The modified specifications also required that the efficiency of *each pump* should be not less than 70 per cent and its capacity not less than 3000 gal. of sea water when lifted to a pressure equivalent to 308.66 lb. To determine whether the requirement was met, a separate test of each pump was required.

58 The efficiency of the pumps was computed by dividing the horse-power output of the pumps by the horse-power input as received from the motors. The horse-power input was computed as follows:

$$\text{h.p. input} = \frac{\text{total watts}}{746} \times \text{efficiency of motors (93.2 per cent).}$$

The horse-power output was computed as follows:

$$\text{h.p. output} = \frac{\text{wt. per gal. (8.34} \times \text{2.31 head in pounds} \times \text{no. of gal. per min.}}{33{,}000}$$

OLIVER STATION TEST

59 The test of the South Street Station was begun at 12:30 p.m. on September 2, 1908, after about 2 hr. of preliminary running for the purpose of adjusting the delivery pressure; it was continued without interruption for 24 hr. With the exception of a short stop of motor No. 2 which was shut down from 2:11 to 2:41 a.m., September 3, to remedy a slight defect in the insulation of the field coils no pump was stopped. During the time No. 2 was stopped the pressure on the delivery mains fell to about 300 lb.; during the remainder of the test the pressure was maintained at or above the contract requirement, as will be noted by consulting the last column of Table 1.

60 The average results for each hour for the 24-hr. test of all four motors are given in Table 2. It will be noted that the delivery every hour was largely in excess of the contract requirement of 15,000 gal. The smallest delivery for eight consecutive hours occurred at the last part of the test, when the average capacity, as shown by the readings, was 18,447 gal. per min., and the average efficiency was 72.2 per cent. During this time the average pressure pumped against was 314.5 lb., or an excess of about 6 lb. over contract requirement.

61 It will be noted from the last column of Table 2 that there is considerable variation in the efficiency; that during the first hour the efficiency was less than 70 per cent, whereas during the third, fourth and fifth hours the efficiency exceeded 75 per cent. This variation in efficiency was doubtless caused by variation in the amount of water by-passed from the pressure to the suction side of the pump over the balanced piston and through the bearings, and possibly during the first hour by the discharge of some water through the relief valve which was pumped but not metered. The valves for regulating the differential pressure on the balance pistons were nearly closed during the third, fourth and fifth hours of the South Street Station test, but were opened the normal amount for the remaining portion of the test. The amount of water which may leak for maximum difference of pressure around the balance piston of any pump without passing through the meter could not be accurately determined but was estimated to be in excess of 4 per cent. Hence it appears that slight changes in the opening of the valve controlling the differences of pres-

sure at this piston must materially affect the efficiency. The normal opening of this valve appears to correspond to an efficiency of about 72.5 per cent.

62 During the test of the South Street Station all the bearings ran cool with the exception of those on No. 6 pump, which heated up during the third and fourth hours but were brought to a normal condition without stopping the pump or reducing its load by the application of lubricants and cooling water.

TABLE 1 HOURLY AVERAGE OF READINGS OF DISCHARGE AND INJECTION GAGES ON PUMPS[1]

SOUTH STREET PUMPING STATION, SEPTEMBER 2 AND 3, 1908

Hour	Pump No. 6		Pump No. 4		Pump No. 2		Pump No. 1		Pump No. 3		Average		Net Pressure lb.
	Disc.	Inj.	Disc.	Inj.	Disc.	Inj.	Disc.	Inj.	Disc.	Inj.	Disc.	Inj.	
12:30– 1:15	335.0	21.8	332.3	22.6	329.8	22.5	331.0	24.0	332.9	23.3	332.2	22.8	309.4
1:30– 2:15	347.2	20.9	347.0	22.2	345.5	22.1	343.8	22.6	346.2	22.1	345.9	22.0	323.0
2:30– 3:15	345.0	20.9	343.8	21.8	342.8	22.1	341.0	22.5	343.4	21.8	343.2	21.8	321.4
3:30– 4:00	344.2	20.9	343.8	21.9	340.3	22.3	339.8	22.6	342.2	21.9	342.1	21.9	320.2
4:30– 5:00	341.7	21.1	341.8	21.9	340.8	22.6	339.3	23.1	342.2	22.4	341.2	22.2	319.0
5:30– 6:00	336.2	22.4	336.3	22.9	335.3	23.8	333.3	25.3	335.7	23.6	335.4	23.6	311.8
6:30– 7:00	337.2	24.4	336.8	24.9	334.8	25.3	333.8	26.3	335.7	24.9	335.7	25.2	310.5
7:30– 8:00	339.7	25.1	337.8	25.9	335.8	26.6	334.8	27.3	337.7	25.9	337.2	26.2	311.0
8:30– 9:00	341.2	26.6	339.3	26.9	338.3	27.3	337.3	28.8	338.7	27.1	339.0	27.3	311.7
9:30–10:00	344.7	27.6	342.8	27.9	341.8	28.6	343.3	29.3	344.7	27.9	343.5	28.3	315.2
10:30–11:00	342.2	28.9	341.3	29.1	340.3	29.3	340.8	30.8	342.7	29.6	341.5	29.5	312.0
11:30–12:00	343.7	30.4	344.3	29.9	343.3	30.3	342.3	32.1	343.7	30.6	343.5	30.7	312.8
12:30– 1:00	345.7	30.6	347.3	30.1	347.3	30.8	345.8	32.6	347.9	31.4	346.8	31.1	315.7
1:30– 2:15	334.0	30.9	334.5	30.6*			332.6	33.1	333.9	31.4	333.7	31.5	302.2
2:30– 3:00	332.4	31.1	331.0	30.9			330.8	32.8	332.2	31.6	331.6	31.6	300.0
3:30– 4:00	349.2	31.4	348.3	31.1	346.3	31.6	347.3	33.3	349.2	31.6	348.1	31.8	316.3
4:30– 5:00	347.2	31.1	346.8	30.9	346.3	31.3	345.8	32.8	349.4	31.4	347.1	31.5	315.6
5:30– 6:00	346.7	28.6	346.3	28.4	345.3	29.3	345.3	30.6	348.2	29.4	346.4	29.3	317.1
6:30– 7:00	342.2	27.6	342.3	25.4	340.3	26.1	340.3	27.1	341.2	26.1	341.3	26.5	314.8
7:30– 8:00	332.7	21.6	332.3	22.1	330.3	22.6	329.3	24.1	331.2	22.6	331.2	22.6	308.6
8:30– 9:00	332.7	20.9	332.3	21.4	331.3	22.1	329.3	22.6	331.2	21.6	331.4	21.7	309.7
9:30–10:00	331.7	20.6	332.3	20.9	328.8	21.8	328.3	22.8	331.2	22.1	330.5	21.6	308.9
10:30–11:00	334.2	21.4	336.8	21.6	332.8	22.3	331.8	23.3	336.2	22.4	334.4	22.2	312.2
11:30–12:30	336.0	21.9	338.0	21.9	333.6	22.6	334.0	23.8	337.7	22.6	335.9	22.6	313.3

[1] Readings corrected for error, and to center of pumps.
* Pump No. 2 shut down from 2:11 to 2:41 account of motor.

63 It will be noted from Table 2 that the average results of the 24-hr. test of the South Street Station exceeded the contract requirements in capacity, pressure head and efficiency.

64 The horsepower delivered by the motors during the test averaged for the 24 hr. about 920 or about 15 per cent above rating, without excessive heating.

65 Immediately after the close of the endurance test of 24 hours, a short test was run on each motor separately, which was continued long enough after uniform results were shown to obtain 12 to 15 readings. This test was run for the purpose of ascertaining whether there were deficiencies in any of the individual motors, and to meet the

TABLE 2 COMPUTATION OF PUMP EFFICIENCIES

South Street Pumping Station, September 2 and 3, 1908

Hour Beginning	h.p.m.	Total kw. per hr.	Total h.p. from motors (input)	Gal. per min.	Net pressure lb.	h.p. delivered	Efficiency per cent
12:30 p.m.		3875	4841.4	18334	309.4	3311.6	68.6
1:30	757.0	3851	4811.4	18634	323.9	3523.6	73.2
2:30	755.0	3829	4784.0	19217	321.4	3605.7	75.4
3:30		3819	4771.5	19220	320.2	3592.8	75.3
4:30	755.0	3811	4761.5	19145	319.0	3565.4	75.0
5:30		3837	4794.0	18995	311.8	3457.6	72.1
6:30	755.0	3818	4770.3	18970	310.5	3438.7	72.2
7:30	755.0	3815	4767.5	18980	311.0	3446.0	72.3
8:30	757.0	3863	4826.4	19020	311.7	3461.1	71.6
9:30	756.0	3868	4830.2	19120	315.2	3518.3	72.8
10:30	757.0	3873	4838.9	19095	312.0	3478.1	72.1
11:30	756.5	3859	4821.4	19120	312.8	3491.6	72.4
12:30 a.m.	757.0	3870	4835.2	19175	315.7	3534.1	73.0
1:30	756.5	3672	4587.8	18790	302.0	3315.0	72.5
2:30	757.0	3667	4581.6	18776	300.0	3288.4	71.5
3:30	757.0	3890	4860.2	19190	316.3	3543.5	72.8
4:30	757.5	3891	4861.4	19160	315.6	3530.2	72.6
5:30	754.7	3861	4823.9	19110	317.1	3337.7	73.2
6:30	757.0	3865	4828.9	19005	314.8	3492.7	72.4
7:30	754.7	3706	4630.4	18710	308.6	3370.8	73.0
8:30	745.6	3659	4571.5	18100	309.7	3272.5	71.5
9:30	745.6	3651	4536.5	17890	308.9	3226.2	71.5
10:30	745.0	3619	4521.6	17795	312.2	3243.4	71.8
11:30	747.0	3618	4519.1	17806	313.3	3256.8	71.8
Average	756.1						72.5

Average efficiency, 1st period of 8 hr. = 73.0 per cent.
Average efficiency 2nd period of 8 hr. = 72.3 per cent.
Average efficiency 3rd period of 8 hr. = 72.5 per cent.

No. of cycles per sec. 12:30 p.m. to 6:30 a.m. = 25.5
No. of cycles per sec. 6:30 a.m. to 12:30 p.m. = 25.0

requirements specified in the printed specifications for the work, viz: that each pump should be free from defects, should have a capacity of 3,000 gal. per min. and an efficiency not less than 70 per cent. The results of this test showed that the individual pumps operated singly had an efficiency from 4 per cent to 6 per cent in excess of

the average when operated together, and that the capacity for the specified discharge pressure was considerably in excess of the requirement of the specification. It is, I believe, generally the case that individual centrifugal pumps delivering water into a main singly show a greater efficiency by from 4 per cent to 6 per cent than the same pumps delivering together into a single main, due probably to less loss in eddy currents and friction head, etc.

TABLE 3 HOURLY AVERAGE OF READINGS OF DISCHARGE AND INJECTION GAGES ON PUMPS

GANSEVOORT STREET PUMPING STATION, SEPTEMBER 5 AND 6, 1908

HOUR	PUMP NO. 6		PUMP NO. 4		PUMP NO. 2		PUMP NO. 1		PUMP NO. 3		AVERAGE		NET PRESSURE LB.
	Disc.	Inj.	Disc.	Inj.	Disc.	Inj.	Disc.	Inj.	Disc.	Inj.	Disc.	Inj.	
9:45–10:30	342.4	24.7	342.6	25.6	344.8	24.9	346.9	25.2	345.1	25.3	338.4	25.1	319.3
10:45–11:30	347.4	24.7	347.1	25.7	345.4	25.2	348.4	25.4	346.4	25.2	346.9	25.2	321.7
11:45–12:15	348.9	24.9	348.9	26.2	347.9	25.7	349.4	25.9	347.9	25.9	348.6	25.7	322.9
12:45– 1:15	351.4	25.7	350.9	27.7	350.1	26.7	352.4	26.9	350.4	26.9	351.0	26.8	324.2
1:45– 2:15	352.4	25.7	352.4	27.4	350.9	26.7	353.4	26.7	351.4	26.9	352.1	26.7	325.4
2:45– 3:15	352.9	25.7	352.9	27.7	351.4	26.9	354.9	27.2	351.4	27.2	352.7	26.9	325.8
3:45– 4:15	353.9	26.4	353.9	28.2	353.4	27.7	355.4	27.4	353.4	27.9	354.0	27.5	326.5
4:45– 5:15	354.4	26.9	354.4	28.7	353.4	27.9	355.9	28.2	353.4	28.7	354.3	28.1	326.2
5:45– 6:15	349.9	27.4	349.9	29.7	350.4	28.7	351.9	28.9	350.9	29.2	350.6	28.8	321.8
6:45– 7:15	349.9	28.4	348.9	30.2	349.4	29.2	351.4	29.2	349.4	29.4	349.8	29.3	320.5
7:45– 8:15	351.9	28.9	350.9	30.7	350.4	29.7	352.4	29.2	349.9	29.9	351.1	29.7	321.4
8:45– 9:15	351.9	29.7	351.9	30.9	350.6	29.7	353.4	29.9	350.9	30.7	351.7	30.2	321.5
9:45–10:15	352.4	29.7	354.4	30.9	351.4	30.2	353.4	30.4	352.9	30.9	352.9	30.4	322.5
10:45–11:15	354.4	30.4	353.9	31.2	353.4	30.2	354.9	30.9	353.4	31.2	354.0	30.8	323.2
11:45–12:15	354.4	31.2	353.4	31.7	352.9	31.2	353.9	31.7	352.9	31.2	353.5	31.4	322.1
12:45– 1:15	353.9	31.2	352.4	31.7	352.9	31.2	354.9	31.7	353.4	31.2	353.5	31.4	322.1
1:45– 2:15	352.4	31.2	351.9	32.2	352.9	31.2	354.4	31.9	353.4	31.2	353.0	31.5	321.5
2:45– 3:15	350.4	31.7	349.4	32.2	349.4	31.4	351.9	32.2	350.4	31.7	350.3	31.8	318.5
3:45– 4:15	350.9	32.2	350.4	32.7	348.9	31.2	351.4	32.7	348.4	31.4	350.0	32.0	318.0
4:45– 5:15	350.9	31.9	350.4	32.4	350.4	31.2	352.9	32.2	349.9	31.2	350.9	31.8	319.1
5:45– 6:15	352.4	31.2	351.9	32.2	352.4	30.7	353.4	31.7	351.9	31.4	352.4	31.4	321.0
6:45– 7:15	350.9	29.9	349.9	31.2	349.4	29.9	351.9	30.7	349.4	30.9	350.3	30.5	319.8
7:45– 8:15	349.4	28.7	349.4	29.7	347.4	29.4	349.9	29.2	347.9	29.4	348.8	29.3	319.5
8:45– 9:45	348.1	27.4	347.7	28.9	346.4	28.0	348.4	28.0	346.4	28.5	347.4	28.2	319.2

Readings corrected for error of gage and to center of pumps.

GANSEVOORT STATION TEST

66 The endurance test of the Gansevoort station with all the pumps in operation was begun at 9:45 a.m., September 5, after the pumps had been operated for about 20 min. giving uniform results. The test was continued for 24 hr. The method of testing and the

TABLE 4 COMPUTATION OF PUMP EFFICIENCIES

GANSEVOORT STREET PUMPING STATION, SEPTEMBER 5 AND 6, 1908

Hour Beginning	h.p.m.	Total kw. per hr.	Total h.p. from motors (input)	Gal. per min.	Net pressure lb.	h.p. delivered	Efficiency per cent
9:45 a.m.	740	3671	4586.5	17107	319.3	3188.9	69.5
10:45	749	3589	4484.2	17310	321.7	3251.0	72.5
11:45	750	3591	4486.7	17290	322.9	3259.3	72.9
12:45	752	3591	4486.7	17280	324.2	3270.5	72.9
1:45 p.m.	752	3604	4502.9	17285	325.4	3283.6	72.9
2:45	753	3604	4502.9	17315	325.8	3293.3	73.3
3:45	753	3630	4535.3	17345	326.5	3306.1	72.9
4:45	753	3685	4604.0	17670	326.2	3365.0	73.1
5:45	756	3696	4617.9	17855	321.8	3354.4	72.6
6:45	756	3661	4574.0	17825	320.5	3335.2	73.4
7:45	754	3676	4592.9	17775	321.4	3335.2	73.3
8:45	753	3685	4604.0	17755	321.5	3332.5	72.8
9:45	755	3657	4569.0	17720	322.5	3336.2	72.9
10:45	755	3693	4614.2	17755	323.2	3350.1	72.7
11:45	756	3704	4627.9	17830	322.1	3352.8	72.6
12:45 a.m.	756	3753	4689.0	18195	322.1	3421.4	73.0
1:45	756	3760	4697.8	18310	321.5	3436.6	73.3
2:45	756	3735	4665.5	18315	318.5	3405.5	73.0
3:45	755	3725	4654.0	18290	318.0	3395.5	73.0
4:45	756	3743	4677.6	18315	319.1	3411.9	73.0
5:45	756	3784	4727.9	18330	321.0	3435.0	72.7
6:45	755	3747	4681.6	18315	319.8	3419.4	73.0
7:45	755	3723	4656.5	18255	319.5	3405.0	73.1
8:45	755	3722	4655.3	18189	319.2	3389.5	72.8
Average	753.6						72.9

Average efficiency, 1st period of 8 hr. = 72.9 per cent.
Average efficiency, 2d period of 8 hr. = 73.0 per cent.
Average efficiency, 3d period of 8 hr. = 72.9 per cent.

No. of cycles per sec. 9:45 a.m. to 2:45 p.m. = 25.00
No. of cycles per sec. 2.45 p.m. to 4.45 p.m. = 25.25
No. of cycles per sec. 4:45 p.m. to 6:45 p.m. = 25.50
No. of cycles per sec. 6:45 p.m. to 7:45 p.m. = 25.00
No. of cycles per sec. 7:45 p.m. to 9:45 p.m. = 25.25
No. of cycles per sec. 9:45 p.m. to 9:45 a.m. = 25.50

various observers were the same as described in reference to the South Street Station.

67 For the Gansevoort station the efficiency average for 24 hr. was 72.9 per cent, with a variation (excepting the first hour) of less than one-half of 1 per cent. It fell below 70 per cent during the first hour, which was due to the opening of an automatic relief valve on pump No. 2, which discharged some of the water into the suction before it had been metered. For that reason the efficiency during the

first hour has not been considered in determining the performance of the pumps.

68 The least capacity during the eight consecutive hours when all the water pumped passed through the meters occurs from 10.45 a.m. to 6.45 p.m. The average capacity during this time is 17,419 gal. or about 16.5 per cent above that guaranteed. The average net

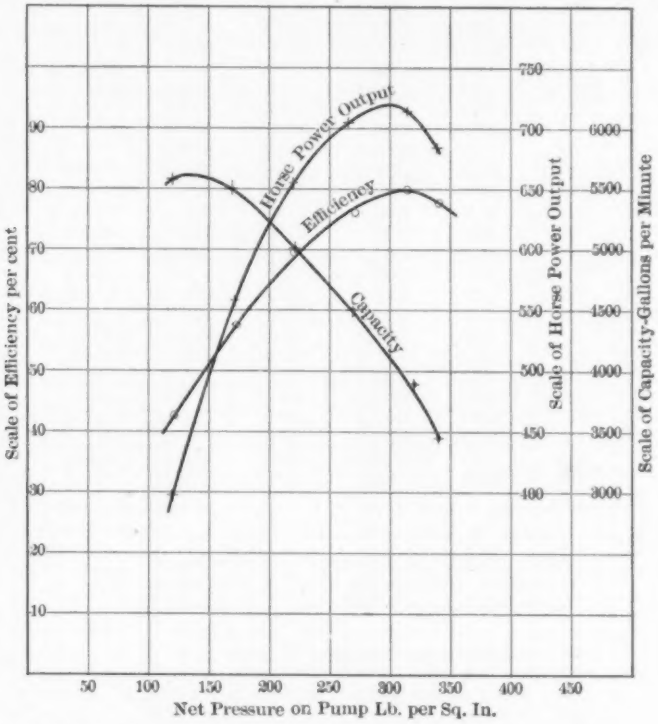

FIG. 7 CHARACTERISTIC CURVES OF THE PUMP FOR VARYING DISCHARGE-PRESSURES

pressure in pounds is 324.3 which is nearly 16 lb. in excess of the contract requirements. The average efficiency for the period above is 72.90 per cent, which is nearly 2 per cent in excess of the efficiency guaranteed.

69 The average capacity for the entire test is 17,867 gal. which was obtained with an average speed of 753.6 r.p.m.

70 Immediately after the completion of the endurance test of 24 hours' duration, each pump was tested when operating alone for a period sufficiently long to obtain 12 to 15 readings after they had

become practically uniform. These tests gave in every case an efficiency several per cent greater than that obtained when the pumps were all discharging into the same main.

CONCLUSIONS

71 It appears from the endurance test in each station that the capacity, efficiency and pressure exceeded the contract requirements by a large margin, and that during the endurance test no mechanical or electrical defects were observed. During the test of the South Street Station one of the pumps was stopped for half an hour to repair the motor insulation, while during the test of the Gansevoort Station no stop of any kind was made. The bearings in both stations were in perfect condition at the end of the test and the temperature of the motors not sufficiently high to interfere with the continuous operation for a longer period. Apparently the endurance test could have been continued indefinitely without injuriously overworking or overloading the pumps and motors.

72 The specifications call for pumping sea water, which most authorities consider to be approximately 2.5 per cent heavier than fresh water. The effect of substituting sea water for fresh water would, for the same horse-power delivered by the motor, have been to reduce the capacity of the pump by about 2½ per cent without sensibly affecting the efficiency. Because of the large capacity over contract requirements shown by each pump, this amount does not materially affect the results.

RESULTS OF TESTS

73 The data and results of the tests at the two stations are given concisely in the following tables. The efficiency is given as computed for each hour, and shows a slight variation which probably can be accounted for by changes in the amount of water leaking past the balancing piston. The individual pump test at the South Street Station shows a variation in efficiency 70 per cent to 77 per cent and at the Gansevoort Station from 76 per cent to 79 per cent. This variation may have been due to the structure of the pumps but in my opinion is more probably due to variable leakage past the balancing piston or through the relief valves.

74 Pump No. 6 at the Gansevoort Station was tested with varying openings of the valve in the discharge pipe. The results are shown in Table No. 6 and also in Fig. 7.

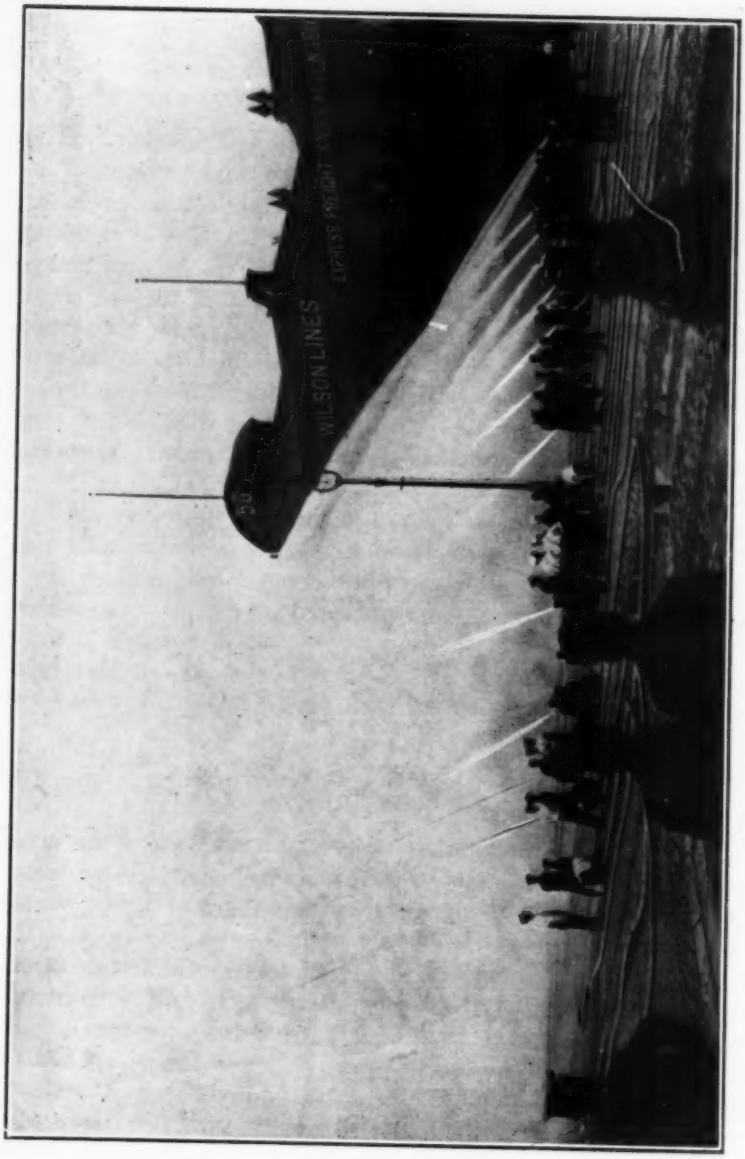

FIG. 8 VIEW SHOWING TEST OF FIRE SYSTEM

TABLE 5 TEST OF INDIVIDUAL PUMPS

OLIVER AND SOUTH STREET STATION, SEPTEMBER 3, 1908

Time	No. of pump	Gal. per min.	Pressure delivery	Lb. per sq. in.		h.p. output	Efficiency of pump
				Inj.	Net		
12:58– 1:14	1	3372	344.4	29.3	315.1	620	74.6
1:22– 1:37	2	3809	336.0	27.9	308.1	683	70.1
1:43– 1:58	3	3495	334.0	28.7	305.3	623	73.2
2:03– 2:18	4	3705	334.5	27.8	306.7	662	76.0
2:24– 2:38	6	3740.7	344.5	28.8	315.7	689	77.0

Immediately following the 24-hr. test for capacity.

TABLE 6 TEST OF INDIVIDUAL PUMPS

GANSEVOORT AND WEST STREET STATIONS, SEPTEMBER 6, 1908

Time	No. of pump	Elect. h.p. input	Gal. per min. Hg. Col.	Pressure delivery	Lb. per inj.	Sq. in. Net	h.p. output	Efficiency of pump
10:05–10:31	1	916	3800	356.8	35.4	321.4	711	77.6
10:36–10:51	2	877	3800	350.8	35.1	315.7	700	70.8
10:54–11:12	3	920.5	3820	350.4	34.1	316.3	703	78.0
11:17–11:30	4	892	3751.4	352.5	35.6	316.9	695	77.7
11:37–11:53	6	899	3880	350.9	35.2	315.7	714	79.4
11:55–12:03	6	880.3	3457	376.1	36.0	340.1	686	77.9
12:03–12:07	6	929	4500	304.4	34.6	269.8	708	76.1
12:09–12:13	6	946	5070	255.6	33.6	222.0	654	69.4
12:24–12:28	6	952	5500	207.4	33.2	174.2	559	58.7
12:32–12:36	6	927	5588	155.2	33.2	122.0	397	42.8

Immediately following the 24-hr. test for capacity.

PRACTICAL RESULTS FROM THE NEW SYSTEM

75 The high-pressure fire system in New York, which was put officially into service on July 6, 1908, has been successfully operated at many fires, but it had a crucial test on January 7, 8 and 9, 1909, when it was brought into service for five simultaneous fires, three of them of more than the usual extent and activity, and one particularly so. Information upon the results attained with the system and the amount of water consumed was given by Chief Engineer I. M. de Varona and published in the Engineering News of Feburary 11, 1909.

76 The fires occurred at Hudson and Franklin Streets, Hester Street and the Bowery, Houston Street and Broadway, Sixth Avenue and 17th Street, and Houston Street and the Bowery. The situation became so dangerous that every engine south of 37th Street, or 40 engines, were summoned, as well as a force consisting of 12

battalion chiefs and more than 600 men, but there was no need to use a single one of the engines.

77 As the violence of the fires increased, additional pumps were brought into service, so that at one time four pumps and motors were in commission at the South Street Station and three pumps at the

TABLE 7 SPECIFIED CHEMICAL ANALYSIS FOR PUMP MATERIALS

	Nickel steel	Medium steel	Steel forging	Steel casting
Parts of 1 per cent.....				
Phosphorus not to exceed.....	0.04	0.10	0.04	0.05
Sulphur not to exceed.....	0.04		0.04	0.05
Tensile strength at rupture, pounds	100,000	650,000	75,000	65,000
Tensile strength at elastic limit, pounds.....	65,000	32,500	38,000	32,000
Per cent elongation in 8 in.....	2	22		
Per cent elongation in 2 in.....	22		22	18
Contraction of area per cent.....	32		32	24
Carbon not less than	20 parts of 1%			
Nickel percentage.....	21 to 24			

Gansevoort Street Station, delivering 35,500 gal. per min. against an average pressure of 225 lb. at the pumps and 205 lb. at the hydrants. During the operation of the pumps 14,095,000 gal. were pumped as recorded by the meters, and the current used was 81,450 kw-hr., the cost of which was $1222.

NOTE.—The high-pressure system was designed by I. M. de Varona, Chief Engineer of the Department of Water Supply, Gas and Electricity of New York. It was also constructed under his supervision. The construction of the electrical machinery was supervised by Prof. Geo. F. Sever as Consulting Engineer. The details of construction were in charge of Thomas J. Bannon, John P. Reynolds and Henry B. Machon, assistant engineers of the department. The machinery of each station was designed and erected by the Allis-Chalmers Co. of Milwaukee.

DISCUSSION

SMALL STEAM TURBINES

By Geo. A. Orrok, Published in The Journal for May

ABSTRACT OF PAPER

The subject of small steam turbines is dealt with from the standpoint of the designing and operating engineer. The various types of turbines now on the market are illustrated by means of sections showing their construction and photographs of the completed machine. A number of steam consumption curves are used to demonstrate the economy that may be expected from the machines in certain situations. The author also describes the details of the turbines, together with some operating conditions.

DISCUSSION AT THE WASHINGTON MEETING

Charles B. Rearick. Small turbines have become popular in large power-house work for driving auxiliaries, and we are furnishing a large number where electric motors and engines of the reciprocating type were formerly used. They are being used extensively for the smallest units in hot-well service for surface condensers, the turbine driving a centrifugal pump which carries the condensed water away from the condenser; supplanting in these cases the usual reciprocating pump. They are more efficient than the reciprocating pump, they take less space, there are fewer parts to maintain, and the service seems to be very popular. They have also proved their worth in larger installations for driving boiler-feed pumps of the multistage turbine type.

2 Some of the newest work taken up by turbine drive is for circulating pumps for surface and jet condenser work. We have a representative lot of such installations using turbines from 50-h.p. to 250-h.p., operating such pumps under low heads and at speeds as low as 600 r.p.m., direct-coupled to the rotor shaft of the turbines.

The paper on Small Steam Turbines was discussed both at the Washington meeting, May 4-7, and at the Boston meeting, June 11, 1909.

3 The question of economy is not touched upon to any great extent in Mr. Orrok's paper, except to give some curves which cover only one condition of service in most cases and cannot very fairly be compared for the different makes. High economy in small turbine units is in many instances of minor importance. Reliability of service is most important of all. Under the very low speeds for driving circulating pumps and similar pump work the economy cannot be especially good; but in nearly all these large power plants the exhaust steam is all utilized in feed-water heaters, and approximately 80 per cent of the heat is returned to the boilers.

4 There is only one class of service in which high economy is absolutely necessary, and that is, when the unit becomes the prime mover or the main unit for a plant. In that case the turbine, both condensing and non-condensing, compares well with the engine; for such work is usually driving dynamos and other high-speed apparatus and the speed can be chosen for the best economy.

5 In Par. 14 I wish to make a correction. The Kerr Turbine Company build also a 30-in. size, and their present patterns are suitable for powers as high as from 600-h.p. to 700-h.p. under proper speed conditions.

W. D. Forbes. Mr. Francis B. Stevens of Hoboken, who died a year ago in his ninety-fifth year, seeing in my shop some small Pelton water wheels, told me that in 1854 he had seen a steam turbine in a candy establishment in New York, which ran 1200 revolutions for some twelve years, with little or no attention. It drove a small fan. Mr. Stevens described the machine as practically the same as a Pelton water wheel, except that the bucket was cut in two, each half being placed on a separate disc, and the steam was led to these two buckets by a "splitter" between them, which of course was stationary. Each bucket was made fast to the disc, which was of course keyed to the shaft.

2 What seems strange to me is, if steam turbines are such excellent things and have been known so long, that they are not more generally used.

R. H. Rice. Mr. Orrok has rendered a valuable service to the Society in describing the construction of the various small steam turbines now on the market and in bringing together the economical results to be expected from such machines under proper conditions.

2 It is interesting to note that all the turbines described are of

the "impulse" or "action" type. The "reaction" turbines are not suitable for small turbine work on account of the complication and expense of their bucket systems, and this fact leads to the conclusion that the great flexibility of the impulse type will render it the ultimate type, superseding the reaction machines entirely for all classes of service. Some prominent builders of reaction turbines abroad are already using either one or two impulse stages at the high-pressure end of the turbine in combination with a reaction section at the low-pressure end; while one builder, who has already passed through this stage of development, has abandoned the reaction turbine altogether in favor of a pure impulse design.

3 The flexibility of impulse turbines is shown by the great range of capacity of the Curtis turbine, which is built in sizes ranging from 5 kw. to 14,000 kw.

4 The author describes the construction of seven different turbines, which may be divided into four classes according to the method of using steam, as follows:

a Single-stage, expanding nozzle, one bucket row, one velocity extraction: De Laval.

b Multistage, conveying nozzle, one bucket row per stage, one velocity extraction per stage: Kerr.

c Single-stage, expanding nozzle, one bucket row, multiple-velocity extraction: Terry, Sturtevant, Bliss, Dake.

d Single or multistage (depending on capacity), two to three bucket rows, multiple velocity extraction but only one per bucket row: Curtis.

5 The value of these various methods of using steam is clearly set forth in the curves presented by the author (Fig. 25 to Fig. 30), giving the water (steam) consumption of the various turbines in Classes *b*, *c* and *d*. In the diagram in Fig. 1 all these curves are drawn to the same scale so that they may be readily compared.

6 It will be seen that with one exception, a very small machine, which suffers somewhat from this fact, the water-rates of all the turbines in Classes *b* and *c* fall rather closely together, while Class *d*, even with a 50-h.p. machine, shows considerably better results. This is much more marked in the case of the 200-h.p. machine. Class *b* is represented by an eight-stage machine of 150-h.p. and shows slightly better results than the machines of Class *c*, particularly at light loads. This result is insignificant as compared with the complication of the large number of wheels, diaphragms, packings, and length of machine,

and this complication is therefore apparently not justified. The reason such multistage machines do not give better water rates is due to a considerable extent to the high frictional losses caused by rotating wheels in a dense atmosphere of steam at comparatively high pressure.

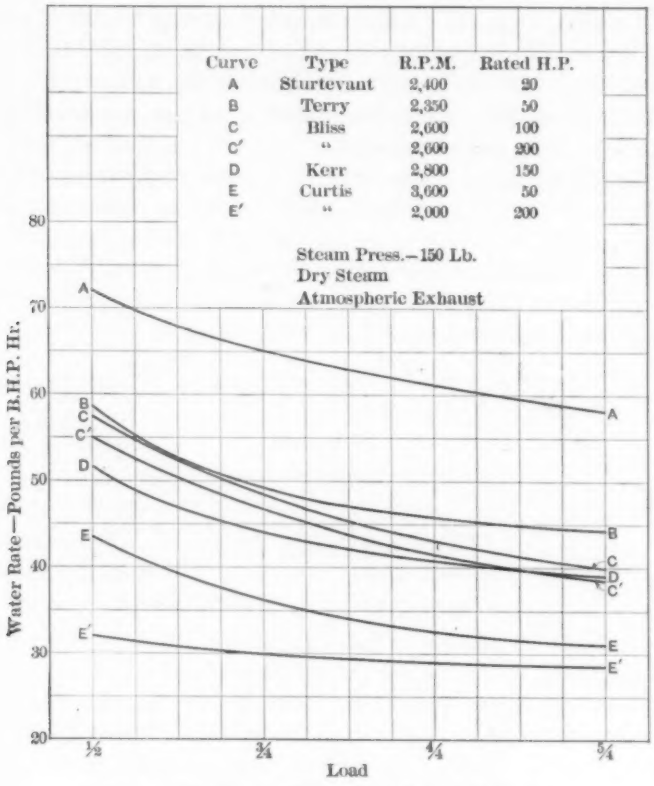

FIG. 1 ECONOMY CURVES OF SMALL TURBINES

7 The turbines of Class *c* labor under two disadvantages, due to using the steam repeatedly in the same buckets; either (as in the case of the Terry turbine) the steam has to be turned at high velocity through an angle of 180 deg. in the return chambers, or it has to be used in buckets which have in general the wrong angle of entrance; for it is easy to see that if the bucket angle is correct for receiving the jet at its highest speed, it cannot be correct when the jet has considerably slowed down.

8 The general principle employed in these turbines was used first by Professors Riedler and Stumpf in the years 1902–1903, and a number of these machines were built and put into service. The company which exploited them, however, abandoned the principle about the latter year and since then has built under a Curtis license, this

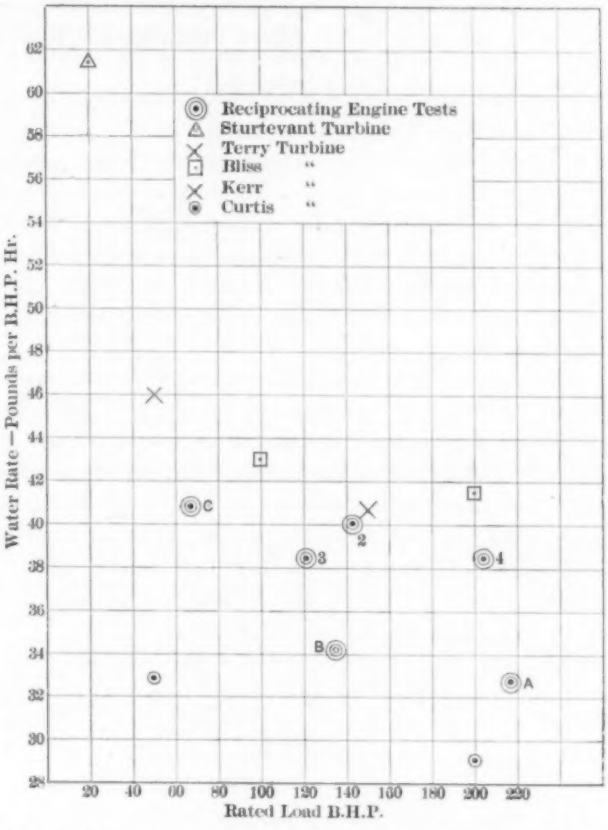

FIG. 2 COMPARISON OF ECONOMY OF RECIPROCATING ENGINES AND TURBINES

step having been taken by reason of the superior economies obtainable by the Curtis construction, which the paper seems fully to confirm.

9 At the Detroit meeting, June 1908, Messrs. Dean and Wood presented a paper giving results of tests on high-speed engines of sizes comparable with the turbine figures given by Mr. Orrok. Fig. 2

shows the results compared with the full-load water rates given by the author.

10 Messrs. Dean and Wood tested engines which had been in service for some time and many of which had evidently seriously deteriorated in efficiency due to wear and leakage. Mr. Orrok confirms the statement made by the writer, that the turbine does not fall off in efficiency after similar length of service and in further confirmation of this is the test on a 75-kw. Curtis turbine made by Professor Carpenter.

11 This fact has also been established by many tests made by the writer on turbines which have been in use for considerable lengths of time, and is subject only to the qualification that when steel bucket constructions are used, as in all the turbines described except the Curtis, wear may be expected under certain conditions of wet steam and light loads which will increase steam consumption after a very moderate length of service. The use of bronze buckets of the proper composition to resist this deterioration is therefore essential to secure the best results under all conditions.

PROF. R. C. CARPENTER. During the past year I have given considerable study to the results from the use of small turbines, arriving at practically the conclusion of the author (Par. 33), that the field of the small turbine is somewhat narrow as compared with the high-speed steam engine. This conclusion applies to small turbines running non-condensing, however; when large turbines are operated condensing the economy is very high, and I think the results will usually be superior to that obtained with piston engines.

2 On testing one turbine, which I think had been in use for three years, I was pleased to find that my results practically agreed with those obtained when the turbine was first installed. This seems to indicate that so far as that turbine is concerned, there was no deterioration from use. The general results which I obtained in the economy tests were substantially those shown on these curves, and indicate that the economy is perhaps not quite as good compared with the piston engine; the advantages of the small steam turbine must therefore be other than simply that of economy. The results of the tests of this machine are shown in the table.

3 I have recently had an opportunity of getting figures from a small turbine operated with a high degree of superheat and running non-condensing. The results of that test were satisfactory in many ways: 350 deg. superheat seemed to have about the same effect as 18

in. of vacuum, and a machine having a water-rate given as approximately 50 lb. per b.h.p. went down to about 22 lb. per b.h.p. The small steam turbine has special advantages for many kinds of work where a high rotative speed and small torque are desirable; for those kinds of work I believe it will ultimately supersede the small piston engine.

4 Mr. Rice made the statement that he believes the reaction

TEST OF CURTIS TURBINE, 75-KW. CAPACITY, OPERATED NON-CONDENSING
October 13, 1908

Test No.	1	2	3
Electric load, kilowatts	57.7	22.6	35.1
Pressure at throttle, pounds gage	121.6	95.0	93.3
Pressure at nozzle, pounds gage	108.4	86.0	82.6
Back pressure, pounds gage	0.24	0.07	−0.25
Barometer reading, inches	30.0	30.0	30.0
Total water to boiler, pounds	14971	1986.5	7454
Wet steam to turbine, pounds	14121	1833	6973
Quality of steam, per cent	98.3	99.0	99.0
Dry steam to turbine, pounds	13881	1815	6903
Dry steam to turbine per hour, pounds	3085	1815	2301
Dry steam to turbine per kw-hr., pounds	53.5	80.4	65.6
Equivalent per i. h.p. (provided 1 kw. = 1.6 i.h.p.)	33.5	50.3	41.0

NOTE—The pressure at the throttle is practically the same as at the boiler, which stood about 80 ft. away.

turbine will drop out of use; but I think he is looking at the question from a prejudiced standpoint. Charles Algernon Parsons, I find, has the opposite view. His notion is that the reaction turbine will replace the impulse turbine for large work. I think there is no question about the advantages of the impulse turbine for small work, but I do not believe the reaction turbine will drop out for large powers when a high vacuum is obtainable.

H. Y. HADEN. A field for small turbines not touched by this excellent paper is that of installations where exhaust steam can be utilized to advantage. There is an installation in Pittsburg of a 150-h.p. turbine connected to centrifugal pumps, which operates under very unusual conditions. Primarily, it takes the exhaust of reciprocating pumps, without any regenerator, and develops the full power when exhausting into a vacuum of 25 in., but it is also capable of automatically taking high-pressure steam at 125-lb. pressure, should the supply of exhaust steam fluctuate too much or be entirely cut off. The same

turbine also operates taking steam at 125-lb. pressure and exhausting into the atmosphere, and it had further to be guaranteed by the manufacturers to take high-pressure steam at 65-lb. pressure when exhausting freely. I believe the above four conditions could not be met by any reciprocating engine. Of course maximum economy will not be attained under each, but it is attained under two: that of exhaust-pressure condensing and high-pressure condensing.

2 The turbine is of the De Laval type, which is particularly adaptable for changing steam conditions, and is the only machine in which any desired plan of operation can be attained by simply changing nozzle ratios, without any change in the angle of the bucket or velocity of

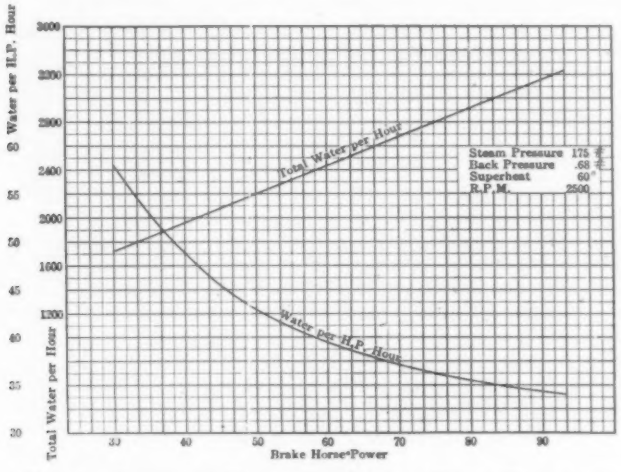

FIG. 1 CURVES OF 90 H. P. AND 50 H. P. TERRY TURBINES

the pump, and without sacrificing either capacity or efficiency. I believe there is a large field for turbines operating under conditions such as the foregoing, whether connected to generators or centrifugal pumps—a field where one can depend upon a unit irrespective of the supply of exhaust steam and without a regenerator or other expensive auxiliary.

FRED. D. HERBERT. Mr. Orrok did not mention tests made on the Terry turbine at the New York Edison Company's plant some years ago, in which the steam consumption is much below that shown in the Terry curve (Fig. 25); possibly because of his preference for tests made outside the builder's works. The accompanying curve and tables show the water rates of 90 h.p., 50 h.p., 25 h.p. and 12 h.p. Terry turbine respectively.

TABLE 1 TESTS OF A 25-H.P. TERRY TURBINE

Test No.	Steam Pres.	Back Pres.	Quality or Superheat	Speed r.p.m.	Load b.h.p.	Per Cent of Rating	Total Water per hr.	Water per h.p. hr.
			Degrees					
1	90	0	62.50	2500	25.52	102.0	1068	41.85
5	90	0	71.70	2500	18.60	74.5	851	45.72
2	90	0	82.42	2500	12.39	49.5	680.5	54.93
10	90	0	50.00	2100	24.29	97.2	1068	44.00
11	90	0	60.26	2100	15.816	63.4	808.25	51.09
6	90	0	45.60	1800	22.90	92.0	1077.25	47.10
7	90	0	79.86	1800	10.21	40.8	638.58	62.54
9	90	0	44.70	1800	15.59	62.4	807	51.76

TABLE 2 TESTS OF A 12-H.P. TERRY TURBINE

Test No.	Steam Pres.	Back Pres.	Quality or Superheat	Speed r.p.m.	Load b.h.p.	Per Cent of Rating	Total Water per hr.	Water per h. p. hr.
			Degrees					
1	136.5	0	58	2501	13.65	113.6	577.3	42.0
2	136.8	0	15	2513	12.08	100.7	547.2	45.3
3	128.5	0	12	2490	11.70	95.6	545.2	46.6
4	130.9	0	0	2501	7.23	60.2	421.5	58.3

2 Regarding the statement that turbines are built of the single stage only, the Terry Company has in operation and under construction several two-stage condensing turbines, the largest of which is 600-h.p. and the smallest about 125-h.p. Because of the successful results obtained, the builders claim that the turbine is superior to the reciprocating engine as an operating proposition, and in none of these turbines now running has a bucket been replaced. No one could make a living on the repairs to small steam turbines, which is one of the disadvantages from the selling point.

W. E. Snyder. Almost all the emphasis has been laid on steam economy. Another point which should receive careful consideration is the lower cost of maintenance, particularly where turbines are used for boiler feed, replacing the direct-acting boiler-feed pumps ordinarily used; for pumping circulating water to condensers; or for driving centrifugal pumps pumping water at comparatively low heads. In all this work the turbine replaces direct acting pumps which are very expensive to operate, not only from the standpoint of steam-consumption, but from the cost of supplies, such as cylinder oil, packing and pump valves. I have in mind one central condensing plant served by a direct-acting pump where the costs are from $75 to $100 a month for packing, cylinder oil and valves. A similar condensing plant served by a small turbine involved practically no expense for these supplies.

2 The first question, three or four years ago, before our company had installed any of these machines, was not a question of steam economy so much as of reliability of operation. If we put in a turbine to pump water for a central condensing plant, into which exhaust a large number of steam engines of various kinds and used for varied service, would the turbine break down, just when it was most needed? For the purpose of investigating this point I went to a plant which had a small unit in operation driving a generator and which had been in use for about four years. The gears showed no appreciable wear, and there had been practically no shut downs.

3 The result of that investigation was the adoption of turbines in central condensing plants in a number of works with which I have to do, and also later for boiler feed and for pumping water under low heads; and they have proved generally reliable regardless of make. This is also true in regard to small turbine air-compressor units for the cupola, etc., in steel works.

4 I think it is, therefore, in the displacement of the direct-acting pump, always expensive to operate, apart from steam consumption, that the small turbine will find its greatest field of usefulness. In the 39th Street Edison Station in New York, the turbine-driven boiler-feed pumps have been continuously operating for months, running almost automatically, and requiring practically no attention or supplies. Steam economies are of course important, yet in large plants where all of the large units are condensing, the steam from the auxiliaries is needed to heat the feed-water, and a few per cent more or less in steam consumption of turbine auxiliaries does not materially change the total economy of the plant.

5 Other advantages in favor of the small turbine as compared to direct-acting pumps, are the small space required and the fact that they can always be kept clean and present a good appearance. Direct-acting pumps are usually very difficult to keep in presentable condition on account of water leakage and of the excessive use of lubricants. The turbine and pump are entirely enclosed, the case can be wiped very conveniently, and it presents nothing of the unsightly appearance which is so often characteristic of the direct-acting pump.

WILLIAM T. DONNELLY expressed the opinion (in response to a question by Mr. Forbes as to why turbines did not come into more general use at an earlier date) that we do not thoroughly appreciate the important bearing which the interchange of knowledge has in the

development of apparatus such as a turbine. The steam turbine has been brought to the attention of people and results have been accomplished with it in recent years because of the general interchange of knowledge and the discussion of engineers on the subject. In the early days of circular sawmills, many of them were driven by 3team turbines. When they were finally replaced by reciprocating engines, there was considerable objection on the score of steam consumption.

2 The development of the modern steam turbine was brought about substantially by the individual work of one man, Mr. Parsons, in persisting in his efforts to develop a turbine which would give practical results. No attention was paid to his efforts for a period of eight or ten years; when he accomplished practical results, and the knowledge was distributed through the engineering societies in this country, we commenced to look into it, and the energy of British engineers in following up the work of Mr. Parsons is the reason why the steam turbine has come to the front. They forced our attention to the marine turbine by building a small boat and accomplishing results. The same was true in regard to the application of steam turbines in driving electric units. I think we can credit the great development of the steam turbine in recent years largely to discussions of technical societies by which the attention of engineers has been brought to it.

F. H. Ball. The conclusions of the author regarding the future of the small steam turbine may fairly be questioned. On the score of efficiency the showing made by the several types, even when tested by the parties interested in their success, is very poor. From these test figures, it appears that the best performance ranges from about 30 lb. per h.p. per hr. to nearly 70 lb.

2 It must be noted also that very high steam pressure is generally used, and in some cases superheat. Under these conditions any good reciprocating engine, even of the single-valve type, run as a non-condensing compound, would easily develop power on 20 lb. or less: the user of one of the non-condensing turbines described must therefore expect to increase his coal bill from 50 to 200 per cent over a single-valve non-condensing compound engine having the simplest possible form of valve gear.

3 Moreover, the speed of these turbines, from 2000 to 3600 r.p.m., will generally be considered objectionably high. Buyers of electric motors generally prefer motors of moderate speed, even at the extra cost, and generally speeds above 1000 r.p.m., even for motors as small

as 10 h.p., are considered objectionable. This same objection must inevitably be urged against speeds of 2000 to 3600 for engines of considerable power.

CHAS. A. HOWARD spoke in reply to the same question, calling attention to the fact that as it is only in the last few years that there has been high-speed machinery for turbines to drive, engineers have been rather afraid of the use of reducing-gearing.

2 As far as a comparison of the merits of different turbines goes, it must be remembered that the economy is affected by the bucket speed even more than by steam pressure. In all of these tests the bucket speeds are different, and any attempt to make a comparison of the steam economy, as in the diagram by Mr. Rice, would thus lead only to an erroneous view. While his curves show in general what may be expected from turbines of this size, no correct comparisons can be drawn between individual machines.

3 It is to be noted also that some of these tests are made by the manufacturers, and naturally under the most favorable conditions possible, while others are made by outside parties with the conditions as they happened to be. This fact, together with the difference in bucket speeds, makes an attempted comparison very deceptive.

4 In comparing the steam consumption of the small turbines with what has been obtained by reciprocating engines under similar conditions, attention is called to the paper presented at the spring 1908 meeting of the Society by Messrs. F. W. Dean and A. C. Wood (Proceedings, June 1908). This paper gives results of tests on several makes of reciprocating engines in which the steam consumption per brake-h.p. hour runs from 39 to 42 lb., which is no better and in many cases not as good as the performance of the turbines.

DISCUSSION AT BOSTON MEETING

DR. L. C. LOEWENSTEIN[1] presented a number of lantern slides showing various types of turbines and discussed the relative advantages of each type. For constructive reasons the peripheral speed of the buckets of the DeLaval turbine cannot be made high enough to obtain the highest efficiency. To reduce the peripheral speed of the turbine and still use the energy of the steam economically, the steam should pass through the buckets a number of times, each row abstracting only a part of the energy.

[1] General Electric Co., West Lynn, Mass.

2 The Kerr turbine is based on the Rateau principle. In an 8-stage Kerr turbine, operating non-condensing with 175-lb. initial pressure, in each stage the pressure would be as follows:

1st	stage	144	lb.	absolute
2d	"	108	"	"
3d	"	80	"	"
4th	"	58	"	"
5th	"	42	"	"
6th	"	30	"	"
7th	"	21	"	"
8th	"	15	"	"

The turbine casing is thus subjected to widely varying pressures. The rotation losses of the eight rotating wheels and buckets may be determined by Stodola's formula

$$\text{Loss in horsepower} = \left(\frac{u}{100}\right)^3 y\,[0.072\,D^{2.5} - 0.6L^{1.25}]$$

in which y is the density of the steam in which the wheel rotates, D is the diameter of the disc in inches, L is the height of the buckets in inches and u is the peripheral velocity in feet per second. The large number of wheels and buckets, besides making a constructive disadvantage, increases the length of the steam path and causes large steam friction losses.

3 The principle involved in the Terry, Sturtevant, Bliss and Dake turbines is that employed first in the Riedler-Stumpf type, and consists in passing the steam through the same buckets a number of times. The disadvantages of this type are, first, the steam must turn through 180 deg. to re-enter the buckets, causing a high friction loss; secondly, the reversing chambers are not constructed to guide the steam correctly into the buckets; thirdly, in some machines the steam re-enters on the same side as it left the buckets, so that if the angles are correct for the steam leaving the buckets they cannot be correct for the re-entering steam. On account of the high water-rate, the principle was abandoned for the large sizes of machines. It is still used in smaller sizes chiefly because of lower cost of construction.

4 The fourth type discussed by Dr. Loewenstein was the Curtis turbine, the illustrations showing a small size of one stage and a larger size of two stages. Curves of steam consumption were pre-

sented to show the comparative water rates of the various classes of the Curtis type. Among the advantages cited for the Curtis turbine were: high velocity of steam through the several rows of buckets; the steam impinges at the proper angle on each row of buckets; buckets of a special extruded metal are used to avoid erosion by wet steam; and an emergency governor prevents excessive speeds.

W. J. A. LONDON.[1] With reference to Richard H. Rice's comparison at Washington of Fig. 25 and Fig. 28, showing the steam consumption of the Terry turbine and that of the Curtis turbine, if the curve of the Curtis turbine be produced and the peripheral speed of the two types be made the same, the curve of the Terry turbine will cross that of the Curtis type at about 1950 r.p.m. See Fig. 1. There

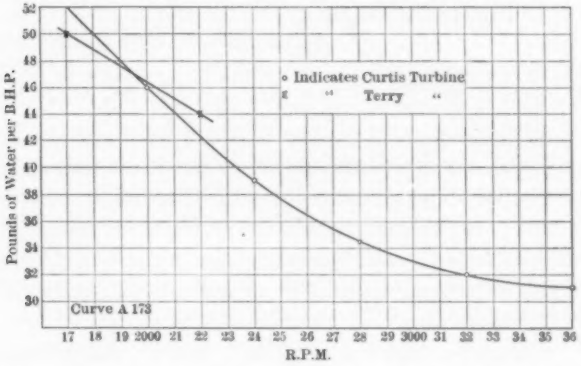

FIG. 1. COMPARISON OF TESTS IN FIG. 25 AND FIG. 28 REDUCED TO THE SAME WHEEL VELOCITY.

is therefore not much room for discussion of the difference of efficiency of the two types. Moreover, with a large turbine an increase of a pound on the steam consumption would increase the cost bill from $2000 to $4000 a year, but with a small turbine it would mean an increase of only from $10 to $25 a year, which would be offset by the difference in first cost.

2 The greatest value of the paper lies in the fact that, better than a salesman, it shows to men having reciprocating engines, the great simplicity of construction of the turbine. Men familiar with reciprocating engines know what to do in case of breakdown, but with a

[1] Terry Steam Turbine Co., Hartford, Conn.

turbine a failure means a shutdown for several days. In a few years, however, every engineer will thoroughly understand the construction of a steam turbine and will be able to make his own repairs. Another fact which hindered the more frequent use of the steam turbine was that generators, pumps, blowers and other machines had to be designed specially for operation with turbines. That this is now being done is an acknowledgment that the steam turbine is here to stay.

CHAS. B. REARICK. The fact that the small steam turbine has obtained a footing is evidenced by the number of installations. It is now used for a variety of pumping service, such as driving hot-well pumps and circulating pumps for condensers; multi-stage pumps for high-pressure work, including boiler feed service; water-works pumping service where maximum economies, obtained only with fly-wheel types of reciprocating pumps of expensive design, are not essential; and for house-service pumps in buildings.

2 The matter of speed is often a compromise, as the pump speeds are not always ideal for the turbine. Especially is this true for circulating work where the heads are often only 15 or 20 ft. and the delivery 20,000 gal. or more per min. It may be necessary to sacrifice some efficiency of the pump in order to run at a speed suited to the turbine. In such instances all the exhaust steam may be used to advantage in heating feed water, while the low cost of operation and the saving in oil and supplies will overcome the cost of increased steam consumption.

3 The steam turbine has made very rapid strides in the driving of blowers for forced draught and such work in large power plants. Many engineers are adopting turbines for driving practically all auxiliaries in large power plants. The question of efficiency is secondary to a great extent for the reasons cited above. It does not matter if the steam consumption of the auxiliaries runs up to 70 lb. per horsepower per hour as a large part of the heat in the exhaust is returned through the heaters. When efficiency is of importance, as in isolated dynamo work, the speeds are usually such as to give results quite as good as those of the reciprocating engine, and in many cases better. The turbine has demonstrated its ability to give economical results in all cases where the speeds are favorable.

4 Regarding the rating of turbines, there is only one point of maximum efficiency of any design, so far as I know, and that is its maximum load. It is very similar to a gas engine in that respect.

If we want maximum efficiency the turbine must carry its maximum load. That can be brought about in some cases by the nozzle system of design in which better economy is obtained at light loads by shutting off nozzles. But where this is done by hand regulation there is always danger of the load coming on without notice or without the engineer having opened any of the hand adjustments. Turbines should therefore be designed to eliminate hand regulation, and to accomplish this some builders provide for automatic operation of these valves. While this is successful within a limited degree, these valves may become leaky in service, and when once leaky the control of the turbine is beyond the operator, which may result in over-speeding to a bursting point in case the load is suddenly thrown off.

5 It follows that the number of controlling valves should be reduced to a minimum. If there is only one valve to control in a machine there is but one valve to look after and to keep tight. Few operators of turbines appreciate how serious the leakage of the controlling valve is to the proper governing of the turbine, especially on very light loads or no load. On the other hand, if turbines are kept well loaded these leaky valves are not noticed and as a result steam consumption is increased through their use rather than diminished, unless all the valves are wide open and the turbine is working up to its full capacity. The turbine which eliminates these dangers, it would seem, is the better machine, and in small units the difference in economy is entirely outweighed by the complications and dangers above cited.

F. B. Dowst. The B. F. Sturtevant Company have for years built reciprocating engines—single engines, multiple-single engines, and multistage engines. Later we built direct-current and alternating-current motors. We first came into touch with the steam turbine principle in 1883 when our attention was first called to the Wise steam motor. This motor was an impulse wheel, with four jets, I think, the steam impinging on buckets, no endeavor being made to expand the steam in nozzles. One of our engineers left us at that time to exploit the Wise steam motor. He returned after a year's sad experience in connection with the amount of steam that would flow through a small opening.

2 Our next experience was with the Dow steam motor. Mr. Dow came to us early in the nineties, I think it was, backed by Mr. Chisholm of the Chisholm Shovel Works of Cleveland, O. This turbine had previously been developed and used successfully to drive

the flywheel in the Howell torpedo. Associated with Mr. Dow was Mr. Howard, for some time connected with the Fore River Ship and Engine Company, as it was then called. The Dow turbine was built in the Sturtevant works and was really a meritorious machine. It was what might be called an inward-flow reaction turbine. A motor of bronze was built and the method which Mr. Dow developed is now used in balancing our rotors.

3 A little later, during the development of the Curtis turbine, a representative of Mr. Curtis came to us for journals for high-speed work. He had trouble in finding a journal box capable of withstanding the high speed of his rotor shaft. A box that we used was very successful in solving the problem. We knew there was a somewhat limited field for high-speed motors for use with our fans, however, and were not quite ready to take up the Curtis turbine commercially.

4 Later we built a turbine for a man from California; I think his name was Hewson. We never reached the testing stage, however, as the machine was burned in our fire of 1901 and Mr. Hewson disappeared so far as we were concerned.

5 A completed Dow turbine was frequently connected with one of our No. 6 blowers. The governing device was not developed, but that was not necessary in order to connect the turbine with a fan. I believe this turbine was tested in our works, and afterwards tested at the Massachusetts Institute of Technology for water consumption, which was found to be high. I have always thought that the Dow turbine possessed great possibilities and wondered why someone did not develop it.

6 When we cleaned out Mr. Sturtevant's desk in 1890, we found over 200 patents on rotary engines, showing his natural interest in high-speed motors. Although many rotary engines have been presented to us, we never found it practicable to use any of them.

7 Experiments with a steam turbine of our own for use with our fans resulted in the turbine described as the Sturtevant turbine. Mr. Orrok's description of the rotor is substantially correct, except that he omitted the fact that in the manufacture of this part we use an open-hearth steel forging of the best quality. Among the uses of the turbine are direct connection with generators, with fans for blowing blast furnaces, and with multivane fans for work on shipboard. Four fans with geared connection, recently built for a heating system in the West, have done good work.

8 An interesting problem was recently presented when the Navy Department insisted on fans for forced draft for the new torpedo boats

where oil fuel is to be used. This of course demanded a slow-speed turbine, but we think we have worked out a satisfactory combination by effecting a compromise between the fan and the turbine element. It is interesting to recall that a few years ago the Navy Department did not consider any motive power except a reciprocating engine. Later the electric motor came into use and now many of the new ships are equipped with forced-draft fans driven by electric motors.

9 There is not the slightest doubt that the general type of turbine discussed in Mr. Orrok's paper is here to stay. Engineers like it and engine builders must get ready to furnish turbine engines.

CHAS. B. EDWARDS.[1] We are more particularly interested in the development of the large marine turbine, but our attention has been recently directed to the smaller turbines owing to the Navy Department specifying them for blowers; there is also a possibility of their use for circulating pumps and other auxiliary machinery on board ship. The Navy Department has increased the steam allowance for turbine installations over what it was a few years ago when it was limited to 50 lb. per h.p. hour.

2 The great problem in marine installations, particularly for naval purposes, is that of weight. The navy contracts specify a certain weight of machine and if we exceed that weight we must pay for it at the rate of about $500 a ton. In considering the turbine proposition, therefore, we must look at it not only from the mechanical side but also from the standpoint of weight. One of the difficulties, of course, is that of the exhaust. The weight of piping, fittings, valves, etc., runs up rapidly and it is therefore desirable that turbine auxiliaries should be placed as close to the condenser as possible; and in order to secure economical results it is also desirable to secure a low velocity of exhaust in the pipe lines.

V. F. HOLMES.[2] The DeLaval Company has recently brought out a combination high-and-low-pressure steam turbine. Many plants where condensing water is available have an excess of exhaust steam from auxiliaries and the question has arisen whether a machine could not be devised for this class of service. That would necessitate storing up energy in times of an excess of exhaust steam to carry the machine over the periods of limited exhaust steam, and would involve expense and complications. What is desirable is a machine in which

[1] Chief Engineer, Fore River Shipbuilding Co., Quincy, Mass.
[2] Power Equipment Company, Boston, Mass.

both the exhaust and the live steam can be used economically without regenerators and other heat-storing devices.

2 The DeLaval combination high-and-low-pressure turbine is built with two nozzle compartments, one for high pressure and the other for low pressure. Each compartment is furnished with nozzles having the proper ratio of expansion for the conditions under which they operate. Some of the nozzles are furnished with shut-off valves for regulation under variable conditions.

3 Two steam connections are provided, one for high-pressure steam and the other for low-pressure steam, each connection leading to its own governor valve, which in turn is operated by a separate governor. The operation is entirely automatic, the low-pressure governor being set for a speed slightly higher than that of the high-pressure governor. On the total or partial failure of the low-pressure steam supply the machine will automatically draw from the high-pressure steam supply the steam necessary to make up the deficiency. Also in case of the complete failure of the low-pressure steam supply, the machine will operate on high-pressure steam, and under this condition will give practically the same economy as a high-pressure steam turbine.

4 The combination high-and-low pressure turbine is built for conditions where continuous operation is essential and where the supply of low-pressure steam is intermittent or is apt to fail completely. The regulation when changing from one steam pressure to the other varies from 2 to 3 per cent, this being on an instantaneous change from one condition to the other, such as seldom occurs in actual service. A by-pass valve allows the admission of high-pressure steam into the low-pressure compartment, for operation under full-load conditions non-condensing. This by-pass valve is not automatic, and is simply to enable the machine to carry full load in case of failure of, or during repairs to, the condensing apparatus.

5 Both the low-pressure turbine and the combination high-and-low pressure turbine are built for steam conditions varying from 5 lb. pressure above atmosphere to 10 in. of vacuum at the steam inlet. They are also built for low vacuums for conditions where the temperature of the circulating water or existing condensers prohibits the maintenance of a high vacuum. The steam consumption of the machine varies somewhat with the sizes and operating conditions; the average machine operating with steam at atmospheric pressure and exhausting into a vacuum of 27 in. to 27½ in. will use from 28 lb. to 32 lb. of steam per b.h.p. hour.

6 The DeLaval Company is also building a high-speed, low-pressure turbine particularly adapted for direct connection to centrifugal pumps and blowers. This class of machine is built in both the low-pressure and combination high-and-low pressure types, and consists of the DeLaval wheel direct-connected to the machinery to be driven. On account of the direct connection of the wheel and the elimination of the usual DeLaval reduction the machine can be economically operated only at high speed, and for this reason is not suited to direct-current generator work, but is particularly adapted for high-speed pumping and blower work, such as power-plant auxiliaries, boiler-feed pumps, elevator pumps, etc.

J. S. SCHUMAKER. An error has crept into these figures that I am sure was not intended. That is, the figures given for the economy of the Terry steam turbine were obtained from a turbine with nozzles designed for 100 lb. pressure. But the steam pressure used on the test was, I believe, 150 lb. One other point that may in fairness be brought out is that the Terry turbine tests as offered here were made without representatives of the Terry Steam Turbine Company being present, while in the majority of the other cases cited the tests are shop tests.

PROF. CARLETON A. READ. I am interested from the fuel side of the question in the use of non-condensing turbines in small manufacturing plants of from 75-kw. to 300-kw. capacity, where there is an excess of exhaust that can be used only for feed-water heating and heating the buildings in cold weather. We all agree that it is well not to have oil in the exhaust if the condensation is to return to the boilers, but many plants have a good and cheap water supply and after using as much of their exhaust as possible still have some going to waste. Nearly all of the tests quoted are from the manufacturers and without doubt are correct for the conditions under which they were made, but data as to coal consumption under actual working conditions would be of interest to the man buying an equipment for a small plant.

PROF. IRA N. HOLLIS. One aspect of the subject impresses me as important. The curves of efficiency used for comparing different turbines relate particularly to the thermodynamic efficiency of the machine or the number of pounds of steam per horsepower developed. It seems to me that where a steam turbine is connected with a pump,

such as one used for feeding a boiler or for circulating water in a condenser, the machine ought to be treated as a whole. From this point of view, the number of gallons of water delivered per pound of steam or per pound of coal is an important factor and should be given in every case. Naturally the pressure against which the water is pumped is another factor. Ordinary reciprocating engines driving feed pumps are very uneconomical machines. I have had experience with pumps that used 100 lb. of steam per i.h.p. or even more. However, the efficiency of the pumps as a whole for delivering water into a boiler was never worked out.

2 It may be that the steam turbine is to replace the steam engine for all purposes about a power station, particularly if the high-pressure centrifugal pump can be developed into a highly efficient machine in connection with the turbine. It seems to me, therefore that it would be useful in connection with all tests of turbines used to drive pumps, to give the combined efficiency of the machine as a whole.

PROF. EDW. F. MILLER. In looking through these figures of steam economies it will be noticed that the greater the load the smaller the amount of steam per horsepower. All the turbines I have had to do with would stand considerable overload, in some cases 80 per cent. I would like to know what decides the maker in rating his turbine. Apparently the economy line runs down as the overload goes on. Why not rate the turbine higher and get better economy?

JOHN T. HAWKINS. I was a pretty old engineer when the turbine was born and consequently know little about it except what I have learned by reading and observation. I am not going to try to impart information but I wish to ask a question. It seems to be a well-known fact that with the turbine engine, the higher the load the greater the efficiency within its limits. To what is the fact due that the turbine is more efficient under high load?

RICHARD H. RICE. Just a few words in explanation of why the turbine water rates decrease as the load goes up and of the effect of the various governing mechanisms on this action. The impulse turbine is essentially a turbine of partial admission. In a multi-stage condensing turbine of this type the buckets in the last stage are usually designed of the right height and the nozzles of the right proportion to use the entire circumference of the wheel. In a four-stage machine, the next to the last stage will use perhaps one-half the circumference of the bucket wheel for steam admission. It

could be designed to use all the circumference, but that would involve undue shortening of buckets. In the second stage there is a further shortening of the arc of steam flow, and in the first stage, a still shorter arc is used, perhaps 90 deg.

2 In non-condensing turbines, if we were to attempt to use the entire circumference of the wheel the buckets would be so small that the machine would be impracticable and inefficient. We must therefore use a short arc, decreasing the cost of the governing mechanism and making a reasonable bucket speed possible. It is evident that the bucket speed must depend on the size of the machine and that, in connection with the operating speed, it is the prime consideration in the cost of the machine. It would be a mistake to make a 25-kw. machine with the same bucket speed as a 300-kw. machine, because the former would be so large in diameter and so expensive as to be impracticable.

3 It follows that one reason why the turbine increases in economy as the load increases is that a larger circumference of bucket wheel is used; a smaller percentage of the total power is wasted and therefore efficiency increases. Therefore, if the governing mechanism works by throttling we have this condition: the steam pressure and area of the nozzle system determine the amount of steam that can be used in the turbine. In a machine with nozzles wide open, the latter must be so designed that the turbine will carry maximum load, as otherwise the turbine would shut down at maximum load. It follows that nozzles designed for full pressure at maximum load will be greatly throttled when running with light load, and consequently the efficiency will decrease. Therefore it is advisable to govern the nozzle system in such a way that nozzles can be designed for full boiler pressure. By using a larger or smaller number of nozzles, and hence a larger or smaller arc of wheel, the full economy of the nozzles is obtained and only the proper number of nozzles are open for a given load.

CHAS. H. MANNING. The diagrams confirm the opinion I had formed of the small steam turbine to the effect that it is a steam thief. But that its virtues will outclass its sins I am thoroughly convinced. Recent developments of high-speed centrifugal pumps, fans and generators open a field for the turbine in which it is sure to succeed.

2 A small practical point is that almost all of these small high-

speed machines use the ring oiler, which is in general bad practice. It has a very small contact on the shaft and any small thing will stop its running. Furthermore, the rings frequently break. If for the ordinary ring oiler a chain with a large arc of contact is substituted, preferably a window-cord chain, it will never fail and will bring up ten times as much oil as a ring oiler. While this is a small point, any machine depends more on the perfection of its detail than it does on the theory on which it is built.

C. P. CRISSEY.[1] There is one type of the small turbine to which the author has given scant space; that is, the small condensing machine. While, perhaps, the field is not so wide for this type as it is for small turbines exhausting at or above atmosphere, it cannot be ignored. Practically all marine work requires condensing prime movers, and many small stations use this type. Only one example of a condensing machine is referred to by the author, the results of tests being shown in Fig. 30. It would be a mistake to consider this curve as representative of small condensing turbines.

2 Small turbines as well as large derive great benefit in economy from high vacuum, and a vacuum of 28 in. is easily obtained on small machines of proper design. In a well-designed small turbine the vacuum shows no greater tendency to fall with the increase of load than in large machines. Why the Kerr turbine shows a loss in vacuum as the load and hence steam flow increase, we are unable to tell definitely from this paper. It will be noted that the steam is discharged from the buckets on each side of the wheels. It is therefore necessary for one-half of the total flow to pass about the wheels in order to reach the succeeding nozzles. Excessive velocities and throttling will occur in the low-pressure stages where the volumes encountered are great, unless large areas are provided for this steam. I understand that in the Kerr turbine this throttling is obviated as much as possible by providing holes in the wheels. These holes, however, increase the windage loss.

3 One of the reasons for abandoning the Riedler-Stumpf turbine in Germany was the inability of its buckets to handle large volumes of steam successfully. The same objection holds against all machines of the Riedler-Stumpf type.

4 The only small turbines having buckets capable of caring efficiently for large volumes of low-pressure steam are the De Laval and the Curtis types. The DeLaval turbine is seriously handicapped

[1] General Electric Co., West Lynn, Mass.

by its high rotative speed, while the Curtis turbine, due to its pressure and velocity stages, is capable of moderate speeds. In order to show that the results of Fig. 30 are not typical of all small condensing turbines, I will say that Curtis condensing turbines of from 100 to 200 h.p. give economies of 18.5 to 15.5 lb. of steam per b.h.p. hour when operating with 150 lb. dry steam and 28 in. vacuum.

5 Regarding the curves of this paper, I believe they should be compared at rated speed, because the bucket angles are designed for this speed. The rated speed may be taken as the maximum stated upon the curves.

CHAS. B. BURLEIGH presented a comparison of water-rate curves given in the paper with a view to determining the relative efficiencies of different types of turbines. As the results derived are substantially in accord with those previously given in a discussion upon the paper by Richard H. Rice at the Washington meeting, they are not here published. Originally the Curtis buckets were milled in the wheel peripheries, said Mr. Burleigh, but experience has demonstrated that where any perceptible moisture is present in the steam, steel turbine buckets will wear. The liability of operation under wet steam conditions is much more common in small steam units than with large ones, as superheaters are seldom if ever installed in small plants, pipes are seldom covered and long steam mains are usual. For this reason the use of steel buckets in the Curtis turbine was abandoned some years ago, and all turbines from the smallest to the largest are fitted with composition buckets upon which many years of use with wet steam has no appreciable effect. Mr. Burleigh continued, as follows:

2 The paper states that the Sturtevant, Bliss and Curtis machines are provided with emergency governors. It would be interesting to learn if the Terry, Dake and Kerr are not similarly equipped. I should feel it to be a serious handicap against their successful introduction.

3 In Par. 25, the author emphasizes the reliability of this type of prime mover, which to my mind is the principal justification for its existence. This is greatly enhanced by its simplicity. The only moving parts, other than the shaft, are the wheels; and, like the hydraulic turbine, omitting the bearings, in no part of the structure is moving metal brought in contact with stationary parts. Can anything more simple be conceived?

4 In Par. 22 is the statement that "in none of these machines is

clearance an important factor.'' This is characteristic of the expanding-nozzle turbine regardless of size, and is due to the fact that in this type of machine all steam expanded by the nozzles is at a pressure corresponding to the stage into which it is admitted; therefore the atmosphere surrounding the buckets is of a uniform density and there is no tendency for the admitted steam to change its course and escape into an atmosphere of less density.

5 I wish to comment on the author's implication in Par. 33, that the small turbine is less efficient than the high-speed steam engine, where he says: "The field of the small steam turbine is somewhat narrow when compared with the high-speed steam engine. The small turbine has its place, however, and with the development of a more economical machine at lower speed ranges, will have a much wider field."

6 I will readily admit that its present speed characteristics limit its field in comparison with the high-speed engine to the extent of the mechanical application of its output; but I will not admit that the present efficiency of the Curtis type in any way limits its field in comparison with the high-speed engine, nor do I think the author intended it to be so understood. But to obviate any possibility of error, I will call attention to a paper presented last June by Messrs. Dean and Wood before this Society and the discussion which followed by Messrs. Young and Treat, detailing the results obtained from water-rate tests of some 14 high-speed engines of different design, which had been in service three months or longer. As these water-rates were given on the indicated horsepower and on the kilowatt basis, I have added 5 per cent for friction and 5 per cent to the water-rate in each case, to facilitate a comparison on a brake-horsepower basis in accordance with the curves forming a part of this paper.

7 Mr. Dean's figures are as follows:

Engine	1	2	2	3	4	4
Capacity, h.p	129.5	106	167.4	153.3	167.68	224.7
Water-rate, lb	38	38.5	37.59	33.6	39.37	38.5
Engine	A	A	B	B	C	C
Capacity, h.p	178.8	286	108	144	77.9	116.8
Water-rate, lb	36.4	31	38.1	35.2	38.5	35.7

8 Mr. Young's figures are as follows:

Engine	1	2	3	4	5	6
Capacity, h.p	70	30	28	130	123	45
Water-rate, lb	48.3	39.9	37.8	36.3	32.2	31.9

9 Mr. Treat's figures are as follows:

Engine	1
Capacity, h.p.	30
Water-rate, lb	43

10 In order to compare the foregoing water-rates per brake horsepower with the curves of standard turbines of the Curtis type, as shown by the curves in Fig. 28 and Fig. 29 of this paper, I have tabulated the results, placing the turbine water-rate under the water-rate of the corresponding capacity of engine, and we have:

Horsepower	28	30	30	45	70	77.9	106	108	116.8
Engine water-rate	37.8	39.9	43	31.9	48.3	38.5	38.5	38.1	35.7
Curtis water-rate	41	41	41	32	31	31	30	30	30

Horsepower	129.5	144	153.3	167.4	167.68	178.8	224.7		286
Engine water-rate	38	35.2	33.6	57.59	39.37	36.4	38.5		31
Curtis water-rate	30	30	29	29	29	29	29		29

11 It will be noted from the foregoing that on the smallest sizes it has been necessary to compare the half-load water-rates of the turbine with the full-load water-rates of the engine for the reason that the smallest Curtis curve in the paper is 65 h.p., but even under these conditions the average of the four smaller engines at full load is only 0.85 of a pound better than the half-load water-rate of the turbine.

12 From this point up we have an exact comparison and at no point does the engine water-rate begin to compare with that of the turbine. The nearest approach to it is at 150 h.p., and here the turbine is 5 per cent better. The widest margin is at 160 h.p., where the turbine is 36 per cent better, while the average from 70 to 250 h.p. shows the turbine to be 22 per cent more efficient than the engine.

13 The paper states that these curves were obtained from the manufacturers and apply to new machines, while the engines were tested in service and had been in use for some time. The discrepancy, therefore, between the actual water rates obtained and the builders' guarantees on which it is customary for these engines to be sold, would lead one to infer that the engines deteriorated with use.

14 On the other hand, the Curtis turbine does not fall off in efficiency due to long service, nor is its efficiency affected by adjustments. In this connection I will refer to the statement made by

Professor Carpenter in discussing this paper at the Washington meeting that he had tested a 75-kw. Curtis turbine which had been in service some 7000 hr. and the results were not materially different from those presented by a new machine of the same capacity and design.

W. J. A. LONDON. From Mr. Burleigh's remarks it is apparent that I did not make myself clear in connection with the curve mentioned by me earlier in the discussion; namely, a comparison between Fig. 25 and Fig. 28. In making this curve, I used only absolute tests according to the figures mentioned and made no deductions whatever except in the question of relative peripheral speeds. If the curves plotted in Fig. 28 are reproduced for a series of full-load points on a speed basis, a positive curve will be formed. On this curve is plotted the two full-load tests shown in Fig. 25. Now, then, as there are only two points given in the Terry tests, it is impossible from these tests to obtain the nature of the curve, but the point I particularly wished to bring forward was that these two points practically coincide,—one test, as a matter of fact, being better and one worse,—both of them being so near the Curtis curve as to make little difference. They are not so far away as Mr. Rice would have us believe from his diagram.

2 Mr. Burleigh raised the point as to whether emergency governors were fitted on other makes of turbines besides the Curtis. Particularly with the Terry turbine and I believe with the majority of the other makes, an emergency governor is not provided for the reason that a positive type of governor is fitted on the main shaft. The worst that can happen is the breaking of a spring, which would immediately close the valve. With a governor driven by a gear shaft an emergency governor is more essential for the reason that the gears are likely to break; hence some form of governor is used on the main shaft. If the direct-connected governor on the main shaft is likely to get out of order, why then is the emergency governor not likely to get out of order when placed in the same position? Up to the present time the Terry Turbine Co. has not had a machine burst, and with the type of governor adopted and the speeds employed, the designers consider an emergency governor an unnecessary luxury.

R. H. RICE Mr. London claims that the steam consumption of the Terry turbine is the same as that of the Curtis turbine, when

operating the Terry turbine at its designed speed and reducing the speed of the Curtis turbine to two-thirds of its designed speed of 3600 r.p.m. The inaccuracy of this comparison can be readily understood when it is known that the angles of the buckets in the Curtis turbine would be radically changed if designed to run at two-thirds of the present rated speed.

2 In discussing emergency governors, it must be realized that we are dealing with much higher speeds than those usual with reciprocating engines. It has been found best in many plants to install emergency governors on reciprocating engines. If this is desirable on slow-speed apparatus, how much more desirable, and even necessary, is it on high-speed apparatus like steam turbines. Many other accidents besides the breaking of a spring can happen to a positive type of governor fitted to the main shaft, and any one of these is sufficient, to cause a dangerous increase in speed of the turbine. An emergency governor can be made to possess the utmost certainty and reliability of action, since its function is to shut down a machine and not to regulate its speed.

J. H. LIBBEY. The applications of small steam turbines mentioned by the author, except for driving high-pressure fans, refer to uses with auxiliary apparatus in a central power station. For this purpose a small steam turbine must be considered in competition with a reciprocating engine, and in general the choice will be decided by the following considerations: First cost, attendance required, maintenance and repairs, space, economy and influence on design of power station.

2 *First Cost.* At present, when the service permits operation at speeds approximating those for which the turbine was designed, the cost of the turbine is somewhat lower than that of a corresponding reciprocating engine.

3 *Attendance Required.* The attendance required for a small steam turbine is less than that required for any other type of steam machinery. It approaches very closely that required for an electric motor.

4 *Maintenance and Repairs.* In Par. 32, the author indicates that small turbines have been running for only three years. Obviously in this time no data of great value could be obtained to enable a decision to be made in regard to maintenance and repairs. The evidence, however, strongly indicates that they will be materially less than for a reciprocating engine.

5 *Space.* Steam-turbine-driven apparatus is generally characterized by the small space required. In a great many cases, a turbine unit can be installed where a reciprocating engine would be impossible.

6 *Economy.* An inspection of Fig. 28 and Fig. 29 shows that for the best conditions a turbine can deliver a horsepower with as little steam as, or less steam than, the same size reciprocating engine. In installations where the conditions are not favorable, the economy is reduced. Unfavorable conditions for a steam turbine are low superheat, low steam pressure, high back pressure or reduced speed of the turbine, on account of the characteristics of the driven machine. The last condition is the most likely to cause reduction in the economy.

7 It should be borne in mind that in the ordinary large central station where fairly large generating units are installed, the steam consumption of the auxiliaries does not in general amount to more than 10 per cent of that of the main generating units. In such cases, the auxiliary exhaust will heat the feed water to about 175 or 180 deg. fahr. A considerable increase of steam consumption of the auxiliaries can be permitted before there is sufficient exhaust to heat the feed water to 212 deg. fahr. In most cases, therefore, the steam consumption of these small auxiliaries is a matter of secondary consideration.

8 In the various auxiliaries generally used the inherent conditions which would affect the steam consumption would be in general as follows:

Exciter, favorable.
Circulating pump, speed low for best results.
Hot-well pump, favorable.
Forced or induced-draft fans, speed low for best results; special design of fan required.
Feed pump, speed low; however, the steam consumption of a turbine-driven multistage centrifugal feed pump is much lower than that of a reciprocating pump of the same capacity.

9 In this connection Mr. Orrok's statement in Par. 32, that there seems to be no change in steam use with length of service is of importance as it is well known that the steam consumption of engines or pumps increases greatly with wear of valves, rings, pistons, cylinders, etc.

10 Many central stations are today operating the auxiliaries

with superheated steam. Very few changes are required in the structure of a steam turbine to adapt it to superheated steam by the use of which the economy is improved. The reciprocating engine gains in economy from superheat, but greater changes in the design are required to obtain satisfactory operation, and the expense of the engine is therefore increased.

11 *Influence of Design on Power Station.* Turbine-driven exciters are generally light in weight and compact. They can be set on the engine-room floor without a heavy foundation or resulting vibration.

12 Circulating-pump units are of simple design. In many cases, a combination of auxiliaries may often be effected. There is on the market a jet condenser, the centrifugal pump and air pump of which are on the same shaft with the turbine. When in surface-condenser work the conditions are such that the speed of the circulating pump is subject to little variation, the turbine, circulating pump, and hot-well pump can be mounted on the same shaft. One manufacturer is prepared to add a rotary vacuum pump to these, either direct-connected or chain-driven. This arrangement gives practically one auxiliary for a surface condenser in place of three.

13 *Future Designs.* The small steam turbine has sufficiently justified its existence. The future will undoubtedly show types with improved economy, especially at reduced speeds, simplicity of design, rugged characteristics, ability to operate without attention, interior construction that is easily accessible and such that few repairs due to wear are required.

THE AUTHOR. The author is greatly pleased with the reception accorded his paper and the amount of discussion which it brought out. He must take exception to a comparison of water rates plotted on percentages of load as abscissæ, and a new diagram has been prepared, Fig. 1, showing the results of all the water-rate curves plotted with bucket speed or peripheral velocity as abscissæ, obviously a much better measure of the performance of these machines. The author would like to take up the question of improper entrance and discharge bucket angles in machines of the Riedler-Stumpf type, as well as the fluid friction question, both mentioned in the discussion of Mr. Rice; but these should be the subject of a mathematical paper and are not of serious importance in a small turbine. The author agrees with Prof. Carpenter rather than with Mr. Rice as to the future of the "reaction" type of turbine. He feels that Mr. Ball has failed to grasp the fact that with small rotating masses speeds of from 600 to 3000

r.p.m. are not as objectionable as a speed of 150 r.p.m. in a modern four-valve engine, or 100 double strokes per minute in a direct-acting pump.

2 Replying to Prof. Hollis' discussion, the author knows of many power plants where entire reliance is placed on turbine-driven multi-stage centrifugal pumps for feed-water service. He knows of no case where an attempt has been made to find the coal consumption

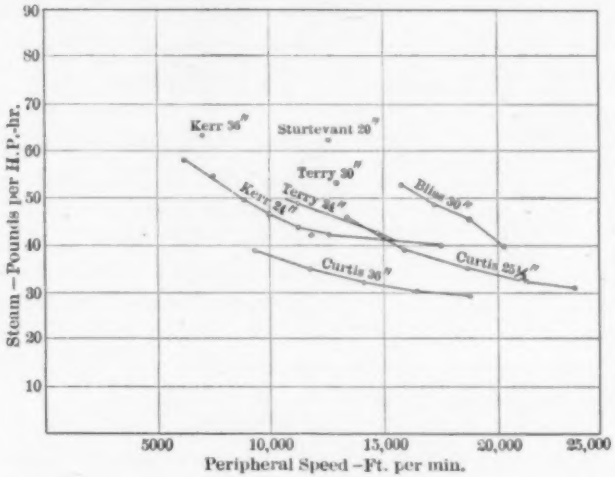

FIG. 1 STEAM CONSUMPTION OF SMALL TURBINES PLOTTED WITH PERIPHERAL SPEEDS AS ABSCISSAE

of the feed pumps directly; in other words, the duty of the pumps. It has usually been obtained through the steam consumption with a knowledge of the evaporation constant of the plant. The use of Venturi meters in the feed lines and in the steam connections to the turbine-driven feed pumps would give this duty directly, and a partial installation of this nature has been made at the Waterside Station of the New York Edison Company.

GAS POWER SECTION

DISCUSSION PRESENTED AT WASHINGTON MEETING

REPORT ON GAS PRODUCER DEVELOPMENT ABROAD

C. L. STRAUB.[1] We have received reports from abroad of progress in the marine gas-producer field, a summary of which I hope will prove of interest.

2 In Holland teaming is practically unknown and local freight trains are never run, the canals being used for moving freight from one city to another or between different parts of the same city. The canal barges range from 40 ft. in length, with an 8-ft. beam, drawing 3 ft. of water, to 150 ft. in length, with a 20-ft. beam, drawing 6 ft. of water. The majority of the barges are hand-propelled, about 9 per cent have steam equipment, while about 6 per cent are propelled by gas engines. Of the last-named, a few—about $\frac{1}{4}$ of 1 per cent—use gas producers, the others using liquid fuel.

3 The gas-engine barges range in size from 40 ft. to 130 ft., the engines from 10-h.p to 300-h.p., and the engine speed from 250 r.p.m. for the larger to 400 r.p.m. for the smaller engines. The reason for the small number of power boats is the great scarcity of fuel. Holland is without coal mines or any natural growth of timber. Hence coal is expensive and difficult to obtain, though wood is more plentiful as large quantities of lumber are shipped in from Germany. Peat is used a great deal, while compressed peat and some domestic and imported briquettes are burned to some extent.

4 The gas-engine boats using liquid fuel are more handicapped by fuel conditions than the steamboats are. The Standard Oil Co. supplies fuel in a few large cities such as Amsterdam and boats of large power can work around these fuel depots only. This limit to the radius of action prevents the extensive equipment of boats with gas engines operating on liquid fuel.

5 The producers installed on the comparatively small number of boats include principally modifications of standard German stationary types. They are not successful and apparently cannot be made to

[1] C. L. Straub, Loomis, Pettibone Co., New York.

give continued good results with the fuels available. Producers that will operate economically on peat, wood, briquettes or poor coal, and require little attention, are in great demand. The engines are giving satisfaction. Boat owners want gas-engine equipments but not for use with liquid fuel.

6 Herr Schichlam, a large ship-builder of Scheidens, could probably be induced to try an American producer, which if successful would result in a large number of orders. There is a great demand for gas-producer equipment of capacities up to 300-h.p. for inland waterway traffic, and for capacities up to 600-h.p. for general inland and foreign traffic. Holland thus offers an attractive field, because success with these small capacities will open the way for larger equipments.

7 It is reported that the Capitaine Company in Germany are making tests of a bituminous coal gas producer, which will continue up to September, 1909. The superintendent of the company stated that although bituminous coal gas producers were used extensively in shore installations, the gas was not pure enough and was not suitable for marine gas engines. A long run cannot be made with any bituminous producer yet designed. Ship conditions are entirely different from those of stationary work.

8 A test run of two tug-boats was made from Hamburg to Kiel and return, one boat having a steam equipment, and the other a gas-producer equipment. The results are shown in Table 1. The weather was rough and the speed maintained 8½ knots an hour. The coal consumption for a period of 10 hours was as follows: Gastug, 530 lb. anthracite coal; Elfrieda, 1820 lb. bituminous coal; an economy in coal consumption of nearly 3.5 to 1 in favor of the gas-producer equipment.

TABLE 1 COMPARISON OF STEAM AND GAS PRODUCER EQUIPMENTS

	Gas	Steam
Name	Gastug	Elfrieda
Length	44 ft. 3 in.	47 ft.
Beam	10 ft. 6 in.	12 ft.
Horsepower	4 cyl., 70 b.h.p	Triple-Exp., 75 b.h.p
Towing meter pull	2140 lb.	2020 lb.

9 In discussing the possibilities of the gas engine and producer for marine use, Herr Flohn, of Germany's largest shipyard, the Stettiner Machinenbau Actien-Gesellschaft Vulcan-Stettin, said that he was opposed to gas-engine propulsion for ships for the following reasons:

a The successful operation of a bituminous producer has not yet been demonstrated to unprejudiced marine engineers.

b The engine will quickly follow the producer when the latter has been perfected, but starting, reversing and cooling must be thoroughly considered. The engines must be designed by marine engineers and not by designers of stationary gas engines.

c The multiplicity of gas-engine cylinders and the large floor space needed are serious disadvantages. The engines are absolutely inaccessible as compared with the modern marine steam engine. An important requirement is that the gas engines shall be double-acting and shall run at the same speed as steam engines of equal capacity.

d Whatever would be saved in smoke stacks and breeching would be more than offset by the large ventilating air duct in the fire room.

10 Herr Blohn, of Blohn & Voss, shipbuilders of Hamburg, said that though his company were using gas to operate a part of their works they had not developed any bituminous producers which they thought suitable for marine use. While there were stationary engineers in Germany who thought they had developed such a producer, marine engineers had not yet seen any outfit which they would endorse.

11 Herr Körting has established a marine department at Kiel, and has practically a monopoly of the government submarine work. He is experimenting and preparing to take up larger work. The German government has decided to adopt gas-engine boats for all warship launches, pinnaces, videttes, portable torpedo boats, and the like.

12 The Nürnberg Company (Vereinigte Machinenfabrik Augsburg und Machinenburg Gesellschaft Nürnberg A.G.) have installed a number of small Diesel oil engines for marine use. Both suction and pressure producers are built by them. As to fuel, the company states that anthracite, coke, charcoal and lignite briquettes are the most suitable for producers. Efforts to gasify raw pit coal have not been entirely successful, at least in medium and small-sized plants.

13 In Great Britain, Vicker Sons & Maxim, Ltd., have built 40,000 h.p. of marine high-speed gas engines. Plans have been drawn for an oil-engine torpedo-boat destroyer of about 30 knots speed. With the same dimensions and speed, the oil-engine destroyer saves enough weight and space so that the armament may be increased from the one 12-lb. and five 6-lb. rapid-fire guns of the steam destroyer to four

33-lb. and two 6-lb. rapid-fire guns. Moreover the quantity of ammunition (number of rounds per gun) is the same for both boats, although the guns and ammunition per round of the oil-engine destroyer are much heavier than those of the steam-engine destroyer. Sufficient fuel capacity is provided to allow a speed of 30 knots an hour with a radius of action 6½ times greater than that of the steam-driven destroyer. Also more space below decks allows superior accommodations for the crew.

14 An outline of a proposed gas-engine propelled battleship is shown in Fig. 1. Here also the advantage in weight over a steam equipment made a heavier armament possible. Each propeller is driven by a 10-cylinder vertical gas engine. The air compressors are

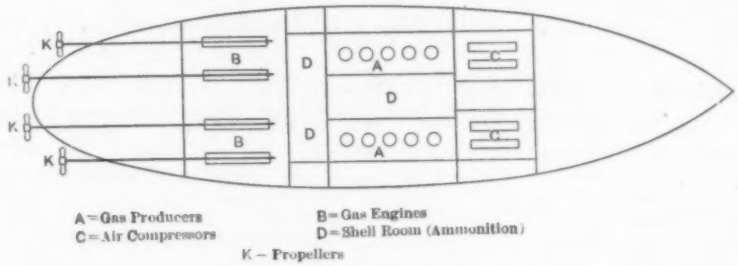

FIG. 1 PROPOSED GAS-ENGINE PROPELLED BATTLESHIP

driven by gas engines. Oil engines drive the generators for supplying current for lighting, and for operating motors driving the steering gear, windlass, pumps, etc. Table 2 and Table 3 show the difference in weight with steam, gas, and oil equipments, and the difference in armament with steam and gas equipments.

15 From a military standpoint the gas equipment shows great superiority in point of magazine location and accessibility. The benefit derived from the absence of boiler breeching and funnels is very marked. Without increasing the length of the ship, the turrets may be so disposed that the ten guns may be fired broadside on either side. A further advantage is that the temperature of the machinery space is lowered and fewer difficulties are involved in the satisfactory heat, isolation and ventilation of the adjacent magazine.

16 A marine gas-producer equipment built by the Beardmore Company under the Capitaine patents was tried out with satisfactory results. The capacity of the plant was 70 h.p., the engine having

four cylinders 8¼ in. in diameter and of 11-in. stroke. The weight of the whole was slightly over 13 440 lb. The equipment was installed in a launch 60 ft. long and of 10-ft. beam. A 10-mile run was made in one hour without recharging the producer.

TABLE 2 COMPARISON OF WEIGHT

	Steam Engine	Gas Machinery	Oil Engine
I.h.p.	16,000	16,000	16,000
Weight of propelling machinery with auxiliaries, tons	1,585	1,105	750
I.h.p. per ton wt. of machinery	10.1	14.48	21.33
Area occupied by machinery, sq. ft.	7,250	5,850	4,110
Area per i.h.p., sq. ft.	0.453	0.366	0.257
Fuel consumption in pounds per i.h.p. per hour, based on experience to date, at full power, pounds	1.6	1.0	0.6
At ¾ power pounds	1.66	1.15	0.75

TABLE 3 COMPARISON OF ARMAMENT

Steam and Gas-Driven Battleships

	Steam	Gas
No. of 12 in. guns	4	10
No. of 10 in. guns	4	—
No. of 6 in. guns	12	—
No. of 4 in. guns	—	18

Note.—The weight of a 10-in. gun is 55 per cent of that of a 12-in. gun.

17 The Beardmore Company have been experimenting with bituminous coal for some time, but have not yet been able to obtain satisfactory results, owing to tar and clinker troubles.

MARINE PRODUCER GAS POWER

By C. L. Straub,[1] Published in The Journal for May

ABSTRACT OF PAPER

This paper treats of the conditions operating against the earlier adoption of producer-gas power for marine service, together with marine gas-power plants in operation at present. All of the present plants are adapted only for anthracite coals or other low volatile fuels and there is a necessity for a good marine bituminous gas-producer plant.

The author compares the up-draft and down-draft system of stationary producer-gas apparatus, showing the relative adaptability of both for marine service. A down-draft plant modified for marine service is also described.

The drawings of a 306-ft. boat fitted with a 1000-h.p. steam plant are given as a comparative example, having been modified to include the installation of a complete producer-gas power plant, with figures showing the space occupied and the weight of the complete apparatus in each case.

The paper states that a 300-h.p. marine producer-gas power plant of this type has been in operation for some time.

DISCUSSION

George Dinkel. I would like to ask the author if there are any gas-producers working with the small grades of the steam sizes of anthracite coal, such as No. 3, 2 and 1 buckwheat, especially as he states in Par. 8 that there are in commercial operation in this country today two distinct types of stationary power gas-producers suited by their design for operation on almost any class of solid fuel. Where are those producers being used on the same grade of steam sizes, and what has been the result?

Henry Penton.[2] As the members probably know, freight is carried on the Great Lakes at a lower cost than anywhere else in the

[1] C. L. Straub, Loomis-Pettibone Co., New York.
[2] Henry Penton, Babcock & Penton, Cleveland, O.

world, and over 75 per cent of the merchant steam tonnage of the United States is built on the Great Lakes. We are, of course, constantly seeking methods of reducing carrying costs, and so far as the ship is concerned the most important item of expense is that of power. We have for some time been firm believers in gas power: power for propulsion, however, is only one of the problems to be met; the problem of the application of gas power to the auxiliary service has given us more concern. At present our ships are handled entirely by steam, both in port and out, and we must have power available at all times.

2 We believed we could depend upon the combustion engineer to perfect the producer gas engine when the opportunity arrived; but it has arrived and while we believe the producer to be satisfactory we are not yet satisfied with the engine.

3 We first considered the installation of gas power in one of our modern ships where the average horsepower (indicated) is in the neighborhood of 2000, but subsequently decided to select a smaller type, believing that we should practice creeping before walking. While the ship to which the author has alluded represents the best standard practice in design, she is not a representative lake steamer in that she is relatively small. Her engines develop approximately 1000 i.h.p. Her carrying capacity is about 4080 tons on 18 ft., while our modern ships run to 12,000 and 13,000 tons on the same draft. The fuel is necessarily bituminous coal, fairly uniform in heat value and averaging about 13,500 B.t.u. Anthracite and coke are out of the question both as to delivery and cost.

4 It should be noted that the fuel consumption given by the author includes fuel used for all purposes aboard ship charged against the actual indicated horsepower. This is the customary method of stating the consumption and is used merely for purposes of comparison. The propelling engines actually do their work on an average of 1.5 lb. to 1.65 lb. per i.h.p. per hr., including the auxiliaries.

5 In Par. 25 mention is made of the installation of a 50-kw. direct current generator. I think, however, it will be necessary to use at least two of this size, depending somewhat on the method adopted for handling the windlass, which calls for the largest individual motor of any of the apparatus. There must be no such thing as a generator shutdown. As just stated, every function of the ship, including propulsion, is now performed by steam, and power must be available every instant from the time the ship goes into commission in the spring until she is laid up the following winter. If we take out steam we must provide something just as available in its place. While a great

part of the time the output will be small, we must be able to take care of the maximum requirements.

6 It may be wondered why, as stated in Par. 26, the auxiliary boiler is required for heat and for washing down decks. These ships run until well into the winter and when the weather becomes severe they ice up badly, and it is not uncommon to make port with 200 or 300 tons of ice aboard. The quickest method of clearing away is with the hot-water hose. I hope we shall be able to use the exhaust gases for generating steam at sea and thus operate the boiler without the use of coal; but I do not yet know whether this is feasible.

7 With reference to figuring the elimination of cost in two years, the operation is brought about in this way: some of our ships get in more, some not so many, but the average is not far from 20 round trips per year; and taking into consideration the reduction in weights, which means additional revenue-producing cargo capacity; reduced space, which in some trades is also additional cargo capacity; and reduction in fuel, which is both reduced expense and additional cargo, I have succeeded in convincing myself that we can get even in about 1½ seasons; but two seasons is perfectly satisfactory, and you can readily see that the addition of one or more trips in the year will go a long way toward the extinction of that cost.

Irving E. Moultrop. Examination of Table 1 gives some very interesting information. It is rather surprising that the total weight of the complete gas-power plant is so much less than that of the steam plant. Of course, the steam plant has a number of auxiliaries which the gas plant does not require, and these auxiliaries are quite heavy, yet up to the present time, in stationary practice, the total weight of a gas engine has been very much in excess of that of a steam engine for the same power. The two prime movers discussed in this paper operate at the same revolutions per minute, so one would naturally assume that the extra weight in the gas engine would go far toward making up for the saving in weight due to the omitting of a number of steam engine auxiliaries. As the total weight of the gas engine room machinery is only about 60 per cent of that of the steam engine room machinery, one naturally wonders if the factor of safety in the gas engine design has not been reduced to save weight, or if this is not the case, if some special weight-saving features have not been introduced in the gas engine design, which might have been used to equal advantage in he steam engine.

2 Comparing the producer room with the boiler room it is inter-

esting to note that the total grate area of the producers is only about five-sixths of that of the steam boilers. Stationary practice has shown that the best producer results are obtained at a very much lower rate of combustion per square foot of grate than in good steam boiler practice. It would be interesting to know how the engineer of the gas power plant expects to obtain capacity out of its producers when the full consumption per square foot of grate area in the producer will exceed what is considered good practice on the grate of steam boilers.

3 In comparing the total weight, machinery and fuel, in the gas-power plant with the steam plant, and also the total space occupied, it should be noted that in the gas plant the bunker capacity is less than half that in the steam plant.

HERBERT M. WILSON.[1] Perhaps it would not be a breach of confidence for Mr. Straub to tell us something concerning the gas producer for the new non-magnetic vessel of the Carnegie Institute. This vessel is being constructed with as little iron as possible, for use in magnetic surveys; and I understand the gas producer and gas engine were selected for auxiliary power because of the small weight of metal required and the possibility of substituting bronze and other non-magnetic metal for iron and steel. Their vessel is about to be launched, and perhaps Mr. Straub could tell something of the gas producer engine plant which is under construction for actual operation.

E. T. ADAMS. Great changes in the weight required have come about in the past few years. The first designs in any line of manufacture uniformly carry unnecessary weight in the various parts and it is safe to say that the gas engine of today, built with the same factor of safety, would be 25 per cent lighter than any engine of the same horse-power built three years ago.

2 This applies to engines in use for ordinary commercial purposes, as electric lighting or power. In view of this fact the statements of the author on this point are not at all surprising. The increasing use of steel and the modification of design based on experience have led to great reduction in weight of engines for special purposes, such as are here specified.

THE AUTHOR. In reference to Mr. Holmes' remarks on the speed of vessels, taking a given hull and a given power equipment, the craft

[1] U. S. Geological Survey, Washington. D. C.

will go at a certain speed, the character of the equipment notwithstanding. I can answer Mr. Holmes' question better by drawing a comparison, with the able assistance of an article by Capt. A. B. Willits, U. S. N.[1] Mr. Willits states that the floor space occupied by the boilers in the New Hampshire equals 0.33 sq. ft. per h.p., and that the weight per b.h.p. is 110 lb. The power of these boilers is rated at their forced capacity. If we install a producer plant and rate it at its forced capacity at 40 lb. of fuel per square foot of grate, it will occupy 1/10 per square foot per b.h.p., for such a plant as the New Hampshire would require, and weigh 30 lb. per b.h.p.

2 From these figures it is apparent that a marine producer-gas plant can be installed in at least the same space and of certainly not greater weight than a modern marine steam boiler plant. These figures refute Capt. Willits' figures of the Westinghouse gas plant, which occupied 1 sq. ft. per b.h.p., and weighed 28.5 lb. per h.p., and the R. D. Wood producer which occupied 1.84 sq. ft. per b.h p. and weighed 194 lb.

3 Mr. Dinkel asked regarding the small anthracite coals in gas producers. I can refer him to the generator of the R. D. Wood plant at Jersey City, which has been operating at the plant of the Erie Railroad for five or six years on a mixture of No. 1 and No. 2 buckwheat coal. The Lehigh Coal and Navigation Company has installed a gas producer which has operated on rice coal and has been running for almost two years. We have two plants in operation, one at Hartford, Conn., and one near Philadelphia, running on a fine grade of anthracite coal, mixtures of Nos. 1, 2 and 3 buckwheat.

4 Regarding the factor of safety in gas-engine parts, which Mr. Moultrop brought up, the six-cylinder engine in Fig. 1, the double-acting engine, weighs less than 30 lb. per b.h.p. Of course, that was a comparatively high-speed engine and delivered 300 b.h.p. at 320 r.p.m. This 1000 h.p. engine will be fitted with cast-steel parts, in almost every instance where cast iron was used on the steam plant, and this makes for a big reduction in weight at a very slightly increased cost.

5 The producer and equipment for the Carnegie, about which Mr. Wilson asked, is almost finished and will be in the boat when she is launched on May 10. The producer shell is about 6 ft. in diameter and is of copper. The pipe and scrubber are of composition metal, containing no iron or steel. The only steel or iron parts on the pro-

[1] Gas vs. Steam for Marine Motive Power, U. S Naval Inst. Proc., December 1908.

ducer are manganese steel grates, doors and door frames, near the hot portion of the fire in the producer equipment, and this manganese steel is less than one per cent magnetic, when compared to mild steel, so that of it we have been allowed the use of 1500 lb.. The engine will have bronze cylinders and will not be lined with cast iron, as the published reports indicate. We are going to run cast-iron pistons in the bronze cylinders. The cylinders are comparatively so thin and so close to the water jackets, that we anticipate no trouble from deterioration. The only steel or iron parts about the engine will be the cams, rollers and valves. The valves will be of cast iron. The steel cams and rollers will be hardened and ground. On a commercial basis, using terms equivalent to mild steel, as far as magnetic force is concerned, we will have less than 200 lb. total of iron or steel in that vessel. The published reports make further detail unnecessary. The boat will be ready to sail July 1.

OPERATION OF A SMALL PRODUCER-GAS POWER PLANT

By C. W. Obert, Published in The Journal for May

ABSTRACT OF PAPER

This paper presents a general description of a producer-gas power plant, in the new Westchester market building of Swift & Company, Bronx Borough, New York City, in operation for over a year and containing both refrigeration and electric power and lighting services. The conditions are unusual for suction gas apparatus, both the refrigeration and electrical services requiring 24-hr. operation continuously, daily throughout the year. The author outlines the operating and maintenance systems developed for keeping both producers and engines in proper condition, and for insuring absolute certainty of their continuous operation which is there particularly necessary.

DISCUSSION

J. A. Holmes. The success of the small producer plant using anthracite coal is practically assured. Not long since (1905), in visiting a number of these plants in Cologne, Germany, I found a newspaper press that had been operated entirely for more than a year by a small gas-producer plant burning small-sized anthracite coal; one of the larger hotels there had been using such a plant for a longer period with entire satisfaction to supply all its electric light and power; in a large commercial house, electric lamps, elevators and all other machinery connected with the establishment were operated by one of these plants. In each of these cases the producer, engine-driven generators and other equipment in the power room, were all operated by one man, and the plant was regarded as a success in efficiency and economy of labor and fuel. In the United States, also, many producer plants have been using anthracite coal for some years. In our own investigations at the Government testing station, anthracite coal has been regarded as a fuel so simple and so easily regulated that we have done little work on it, turning our attention mainly to the bituminous coal producer problems.

2 In regard to producer work with bituminous coal, we have investigated fuels rather than different types of producers. Using every imaginable grade of bituminous coal and lignite in making short-time tests, we have encountered many difficulties due to a lack of familiarity with the special manipulations required by certain fuels. These difficulties, one being to secure a uniform quality of producer gas, would not be met in using the same fuel year after year. In early work, with the Taylor producer, we could get gas of absolutely uniform character not more than an hour at a time, and the variation in 24 hours was at times from 125 B.t.u. to more than 200 B.t.u. per cu. ft. of gas, these variations being largely due, no doubt, to inexperience in the handling of any special fuel. During the past three years, however, with more experience, the regularity and efficiency of this gas has been greatly increased.

3 Another difficulty, and one not entirely separable as yet, is the slagging or clinkering of the ash in the producer. The ash in certain coals slags more readily than in others; and different ashes slag more readily at different temperatures. One of the greatest needs in producer development at the present time is that of a regular mechanical feed of coal and removal of the ashes which now accumulate in some producers, to be cleaned out after the producer has cooled down. We have sometimes found the slag from certain coals, burned at high temperatures, accumulating irregularly on the brick walls lining the producer, at the rate of 6 in. to 10 in. during a week's run. If mechanical arrangements can be devised, by which the ash may be removed from the base of the producer as regularly as from the base of a boiler, then the use of a double producer will be largely avoided. Decided progress is being made in overcoming this difficulty.

4 Still another line of progress is in the reduction of weight and bulk of the producer making its use possible instead of that of steam boilers for propelling ships. Mr. Straub's paper indicates what is being accomplished along this line. Already the anthracite producer and gas engine have been reduced in size and weight to less than those of the steam boiler and reciprocating engine; and the outlook is hopeful for the producer burning bituminous coal.

JOHN H. NORRIS. I have been connected with the manufacture of gas engines for a number of years, and the principal trouble we have had in the operation of gas engines of any size is to overcome the notion that a gas engine needs no care. At the present time, however, gas engines are running successfully because in most installations they

receive proper attention. I am glad to see put on record the statement that a gas engine installation needs as close attention as a steam engine installation.

WILLIAM A. BOLE. The Westinghouse Machine Company has been working on the gas-producer problem as well as on the gas-engine problem for some time, and now believes itself ready to offer gas producers that will be as practicable and as easily manipulated and capable of as long-continued runs as any boiler plant. A producer plant of 175-b.h.p. capacity has been in operation at our works for practically a year, without pulling down the fires. During that period all sorts of runs have been made, continuous runs at full capacity for ten days or two weeks, and the more ordinary runs in which producer and engine are shut down at night; and the producers have burned not only the comparatively good coals of the Pittsburg district and the better coals of the Pocahontas region, but several of the Western and Southwestern lignites and even peats from New England. The latter have not been so successfully burned, but the success in burning Colorado lignites has been very decided. This producer was shut down and cleaned out, simply by shoveling the ashes out of the water seal, and observations of the condition of the interior walls showed that it might just as well have been operated continuously for five years instead of one, or as many years as the firebrick lining would last. The requirements for continuous performance seem to have been admirably met in this design.

2 This producer is designed for the burning of bituminous coal alone, and resembles a small producer inverted and placed on top of a large one, making a double-zone producer especially adapted for the gasification of bituminous coal without passing tar of any description out of the producer shaft. Apparently the only solid material emitted from the gas is a small amount of lamp-black which is successfully removed by the use of a static or stationary scrubber, and the cleanliness of the gas is proved by the fact that practically all the gas was converted into brake horsepower by being employed in the actual operation of a gas engine, without troublesome deposit of any kind upon the ports or other parts of the gas engine.

3 Whether such a producer would be available for marine purposes I do not know: the only question seems to be whether the motion of the ship would interfere seriously with the descent of the fuel from top to bottom. The producer has been subjected to practically every test, and we believe we are about ready to offer it for both large and small plants.

A METHOD OF IMPROVING THE EFFICIENCY OF GAS ENGINES

By Thos. E. Butterfield,[1] Published in The Journal for April

ABSTRACT OF PAPER

The tendency in gas-engine practice to secure higher efficiency by reducing the clearance and increasing the compression, has led to various expedients to avoid premature ignition and other troubles incident to such practice. The author discusses the method of diluting with an inert gas the charge drawn in during the suction stroke of an Otto cycle engine. He refers to the invention of this method by John Saltar twenty years ago and to experiments subsequently made to show its practical working effects. Calculations are given based upon the dilution of the charge with inert gas and comparative tables showing the effect of such dilution.

DISCUSSION

A. M. Green, Jr. The equation of Par. 20 of this paper is not clear to me in derivation. It seems to me that when the total volume of gas and air is $1+C$, the volume of the clearance gas will be $\frac{15}{14}\,C\,\frac{T_1}{1200}$ and the volume of the air will be $1 + C - \frac{15}{14} \times \frac{T_1}{1200}\,C$, and both will be at the temperature T_1, so that the equation will be

$$\frac{15}{14}\,C\,\frac{T_1}{1200}\,(1200 - T_1) = \left(1 + C - \frac{15}{14}\,C\,\frac{T_1}{1200}\right)(T_1 - 530)$$

from which

$$T_1 = \frac{530\,(1 + C)}{1 + C - \frac{670}{1200}\,C\,\frac{15}{14}}$$

which is the same as that found by Mr. Butterfield in another form.

[1] Chief Engineer, Otto Gas Engine Works, Philadelphia, Pa.

My only question is, the reason for the form of equation 1 of Par. 20, the basis of which is not clear.

WM. O. BARNES. It may be of general interest to show that dilution of the mixture in a gas engine will not be attended by a loss of efficiency provided the speed of combustion is not decreased.

2 Let:

V_1 = the volume of the cylinder and compression space at the end of the suction stroke = unity.
V_2 = the volume of the compression space.
P_1 = the pressure at the end of the suction stroke.
P_2 = the pressure at the end of compression.
P_3 = the explosion pressure.
T_1 = the temperature at the end of suction stroke.
T_2 = the temperature at end of compression.
T_3 = the explosion temperature.
n = the ratio of the specific heats at constant volume and pressure.

3 Assuming adiabatic compression and expansion we have, during the compression,

$$P V^n = P_1 V_1^n = P_1$$

4 The work of compression is

$$\int_{v_2}^{v_1} P\,dV = \int_{v_2}^{v_1} \frac{P_1}{V^n}\,dV = \frac{1}{n-1} P_1 (V_2^{1-n} - 1)$$

and since

$$P_2 = \frac{T_2}{T_1} P_1$$

the work of expansion is

$$\frac{T_2}{T_1} \frac{1}{n-1} P_1 (V_2^{1-n} - 1)$$

the net work of the cycle is

$$W = \frac{T_3 - T_2}{T_1} \frac{1}{n-1} P_1 \left(\frac{1}{V_2^{n-1}} - 1 \right)$$

5 Now,

$$T_2 = T_1 \left(\frac{1}{V_2} \right)^{n-1}$$

and assuming an absolute temperature at the end of the exhaust

stroke of 1200 deg. fahr., and of 530 deg. for the mixture as it enters the cylinder, we may write:

$$V_1 T_1 = 1200\, V_2 + 530\, (V_1 - V_2)$$

$$T_1 = 670\, V_2 + 530$$

$$T_2 = (670\, V_2 + 530) \left(\frac{1}{V_2}\right)^{n-1}$$

Substituting for $\frac{1}{T_2}$ in the expression for net work:

$$W = (T_3 - T_2) \frac{1}{n-1} P_1 \left(\frac{1 - V_2^{n-1}}{670\, V_2 + 530} \right)$$

6 The net work is proportional to the rise in temperature during combustion, $(T_3—T_2)$, which is in turn directly proportional to the amount of fuel in the mixture. Consequently in an ideal engine in which no transfer of heat takes place between the gas and cylinder walls, dilution of the mixture would not affect the efficiency; provided of course the combustion be not rendered imperfect or be retarded. Since the temperature attained by the burning of a mixture is lowered by dilution, such dilution will, under usual conditions of wall temperature, increase the efficiency of an actual engine by lowering the jacket losses.

7 The influence of the degree of compression on the work produced and on the efficiency is measured by the factor:

$$\frac{1 - V_2^{n-1}}{670\, V_2 + 530}$$

Values of this factor corresponding to different degrees of compression are given in the following table (n taken as 1.3), in which the first column gives values for volume of compression space assuming the cylinder volume to be unity:

V	*Factor*	*V*	*Factor*
0.5	0.00022	0.25	0.00049
0.45	0.00026	0.2	0.00058
0.4	0.00030	0.15	0.00069
0.35	0.00035	0.10	0.00084
0.3	0.00041	0.05	0.00105

8 Acceptance of an imaginary limitation is sometimes a most effectual means of retarding progress. The prevalent idea that the

amount of compression is limited by the necessity of avoiding pre-ignition is a case of this kind. The only harm that has ever been caused by pre-ignition from high compression has resulted from the fact that the parts of the engine were not correctly proportioned to permit economical combustion.

9 Assuming adiabatic compression and expansion, *a b c d c e* (Fig. 1) is the form of diagram given by an engine in which pre-ignition is taking place. The net work developed can be computed with the above formula by giving V_2 the value corresponding to the volume at *b*. This work is obviously greater than would have been developed had the compression been limited to a point where there was no danger of self-ignition. Of course in an actual engine, running under these conditions, a certain loss of heat to the cylinder walls will take place during the compression after ignition and the re-expansion to ignition volume, but when pre-ignition takes place, it is an indica-

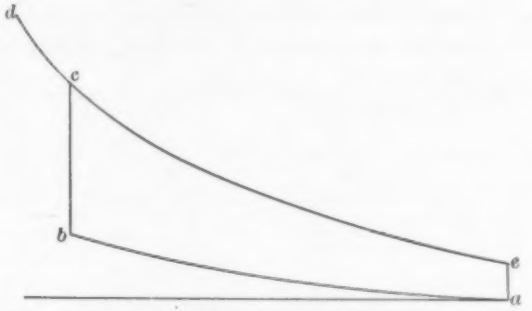

FIG. 1 FORM OF DIAGRAM GIVEN BY AN ENGINE IN WHICH PREIGNITION IS TAKING PLACE

tion that approximately the utmost efficiency is obtained, compatible with the initial temperature and composition of the fuel, and the existing conditions of the cylinder walls.

10 The loud poundings which have sometimes been experienced in attempts to use high compression are not necessary accompaniments of the increased speed of combustion due to such compression, but are simply indications that the designer failed to provide sufficient rigidity of parts and extent of bearing surface. No matter what speed of combustion may be obtained, the gaseous contents of the cylinder cannot deliver a blow or exert an inertia effect on the pis ton of any important magnitude. Hence the question of raising the degree of compression, even to the point of self-ignition, and the com-

plete realization of the gain in efficiency, thereby made possible, is merely a matter of machine design.

11 The desirability of diluting the mixture with inert gas, for the purpose of slowing up combustion, is questionable, especially, since if full advantage be taken of the possible gain in efficiency due to increased compression, it will be found that the first cost of the engine per horsepower will show a decrease due to its superior rating. On the other hand, the dilution of the mixture with air provides a means of regulation, and of running under part-load without loss of efficiency.

THE AUTHOR. The reason the first equation in Par. 20 is in a different form from Mr. Green's corresponding equation is that he used for the basis of his equation the volumes of clearance gas and fresh mixture after the interchange of heat, while I consider them before the interchange of heat. The following derivation of the final equation is better than either form suggested: Since the total volume of gas contained in the cylinder at the end of the suction stroke is $1 + C$, we have at temperature T, by Guy Lussac's law, volume of clearance gas $\frac{15}{14} C \frac{T_1}{1200}$, volume of mixture $(1 + C - \frac{15}{14} C) \frac{T_1}{530}$

2 Then

$$1 + C = \frac{15}{14} C \frac{T_1}{1200} + \left(1 + C - \frac{15C}{14}\right) \frac{T_1}{530}$$

$$T_1 = 530 \frac{(1 + C)}{1 + \frac{675}{1680} C}$$

which is exactly the same as the result first given.

3 All the results given in Mr. Barnes' first seven paragraphs were already fully given in the paper.

4 His statement that the loud pounding caused by pre-ignition in a gasoline engine is due merely to weak construction seems contrary to experience. The trunk piston does not permit enlargement of the piston-pin bearing, and there would be practically no buyers for a small gasoline engine provided with a separate cross-head. Besides this, the principal part of the pound is probably due to elastic deflection of the walls of the cylinder head and piston. The thickness of these walls cannot be indefinitely increased. But an engine operating by self-ignition will be exposed to all the troubles that have been

experienced with hot-tube ignition. The maximum pressures of explosion and the highest available mean effective pressure, would always vary within limits that might lie outside the bounds of safety on the one hand, or of satisfactory development of power on the other hand.

OFFSETTING CYLINDERS IN SINGLE ACTING ENGINES

ABSTRACT OF PAPER

By Prof. Thurston M. Phetteplace, Published in The Journal for February

This paper presents the results of a thorough investigation of the question of offsetting cylinders in single-acting engines which seemed desirable on account of the increasing practice among automobile builders. Curves of the side-pressure of the piston on the cylinder wall, considering the inertia forces due to the weight of the reciprocating parts, are shown for different conditions of engine speed, weight of reciprocating parts, amount of offset, and ratio of length of connecting rod to crank. The maximum side-pressure affecting the design of the piston and the average side-pressure indicating lost work and wear are compared for different cases.

Curves of the inertia due to the weight of the reciprocating parts are plotted and comparative vibration in engines having cylinders offset with those having no offset are discussed.

Finally four definite cases are compared, showing the effect of offsetting on the length of the different parts, on the curves of inertia forces, of side-pressures, and of turning-effort.

The derivation of formulæ are tables and given in an appendix.

DISCUSSIONS

Winslow H. Herschel. In December 1907, I had occasion to investigate the question of offsetting cylinders for large-sized gas engines, and as the conditions are somewhat different from those of the automobile engines considered by Professor Phetteplace, the results may be of interest. I shall consider only the variations in maximum and average pressure on the cylinder walls, since, as indicated in Par. 62 of the paper, these are the main questions at issue.

2 For the sake of simplicity I used the graphical method mentioned in Tolle's Die Regelung der Kraftmaschinen, page 32. The

computations were based on an actual card from a four-stroke cycle producer-gas engine, and upon the following data:

R.p.m. = N = 225
W/A = 4.18
R = 12 in. = 1 ft.
L/R = 5
0.00034 W/AN^2R = 71.8

This last value, 71.8, happens to be very nearly the average of the two corresponding values, 34.4 and 111.38, given by the author in Table 3. Computations were also made using speeds of 450 and 1000, giving inertia constants of 289 and 1430 respectively, but it soon became evident that there could be no gain from offsetting under these conditions, and the investigation was restricted to the speed of 225.

3 As I understand the paper, the author has considered only vertical engines, or horizontal engines where the weight of the reciprocating parts is so small that its direct effect in increasing or decreasing the pressure on the cylinder walls need not be taken into account. In the present case, however, a distinction must be made between vertical and horizontal engines. For the latter, when the side pressure acts downward, due to gas pressure or inertia forces, the weight of the piston must be added, but when the side pressure acts upward, the piston weight is subtracted. It should be noted that for a vertical engine, for a given value of 0.00034 W/AN^2R, that is, for a given inertia constant, as I have called it, it would make no difference whether this value were obtained with a large value of W/A and a small value of N, as in my case, or with a small value of W/A and a large value of N, as in the cases used by the author. But on the other hand it would make considerable difference for a horizontal engine where the value of W enters into the computation apart from the inertia constant.

4 By using the same indicator card as for the four-stroke cicle computations, and disregarding the second and third strokes, I obtained, somewhat approximately, the side pressures for a two-stroke cycle engine.

5 Fig. 1 shows the variation in maximum pressure on cylinder walls, or side pressure, due to variations in offset. The ordinates above the base line ST are proportional to the side pressures. The line AB shows the maximum side pressure for a vertical engine at about the middle of the first stroke. As the offset increases the

angle of the connecting rod for this middle position decreases, while the angle at the end of stroke increases, so that for large offsets the maximum side pressure is found at end of stroke, with values as shown by line *BC*. Similarly for a horizontal engine we get the lines *DE* and *EF*. For the maximum side pressure due to inertia forces we have line *GH* for a vertical, and line *JK* for a horizontal engine.

6 It will be noticed that line *JK* is not parallel to line *GH*. The reason for this is that for the line *JK* we must use the inertia force near the end of the second stroke, which gives a downward pressure on the cylinder walls to be added to the weight of the piston, and this downward pressure is not as large as the upward pressure near the beginning of the second stroke, which was used for the line *GH*.

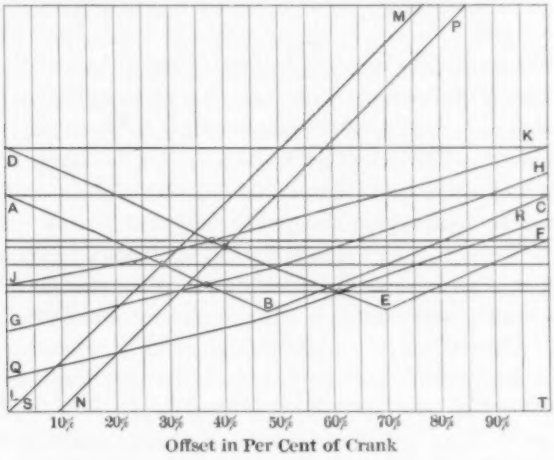

FIG. 1 DIAGRAM SHOWING VARIATION IN MAXIMUM PRESSURE ON CYLINDER WALLS, DUE TO OFFSETTING CYLINDER

7 In the case of a two-stroke cycle engine, where we must use the fourth instead of the second stroke, our maximum inertia force will be near the beginning of the fourth stroke acting upward, so that the weight of the piston must be subtracted, giving line *QR*.

8 If we imagine a somewhat earlier ignition than that shown in Fig. 2 of the paper, the maximum side pressure at or near the beginning of the first stroke will be increased. Whether for this reason or not, I found that with large offsets the maximum side pressure of the first stroke was at the beginning of the stroke, acting upward, with

values as shown by line *LM* for a vertical, and line *NP* for a horizontal engine.

9 Fig. 1 corresponds to Fig. 16 of the paper and may be used in the same way to determine the most favorable offset for the various conditions. Table 1 gives the offset and the per cent reduction in side pressure in each case.

TABLE 1 OFFSET AND PER CENT REDUCTION IN SIDE PRESSURE

Curves	Dominating Factors	Case	% Offset	% Gain
AB and *GH*	Gas middle first, inertia beginning 2nd	Vertical late ignition, 2–4 Cycle	37.0	41.5
AB and *LM*	Gas middle first, gas beginning 1st	Vertical early ignition, 2–4 Cycle	28.7	31.6
DE and *JK*	Gas middle first, inertia end 2nd	Horizontal late ignition, 4 Cycle	37.7	34.7
DE and *QR*	Gas middle first, inertia beginning 4th	Horizontal late ignition, 2 Cycle	61.3	54.3
DE and *JK*	Gas middle first, inertia end 2nd	Horizontal early ignition, 4 Cycle	37.7	34.7
DE and *NP*	Gas middle first, gas beginning 1st	Horizontal early ignition, 2 Cycle	40.6	37.2

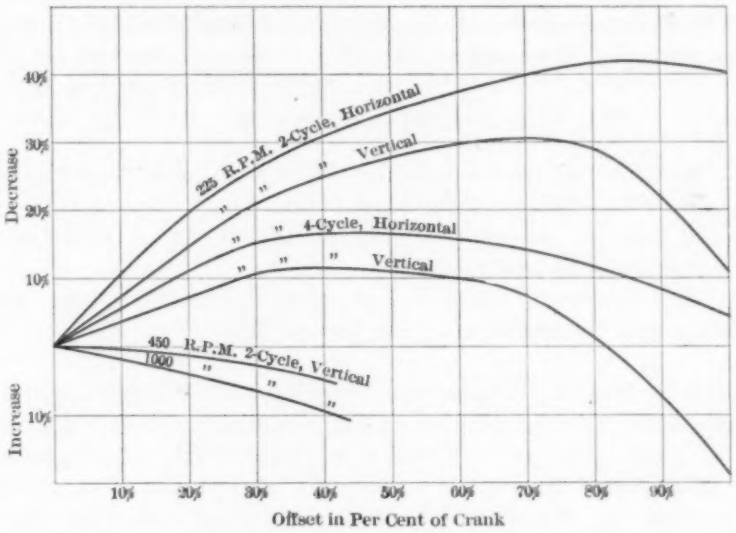

FIG. 2 DIAGRAM SHOWING CHANGE IN AVERAGE PRESSURE ON CYLINDER WALLS DUE TO OFFSETTING CYLINDER

10 The author (Par. 22) finds the most favorable offset to be 50 per cent of the crank length for slow speed, and in Par. 24 and Par. 25 he finds it to be about 20 per cent for high speed. These

values may be compared with the first line of Table 1; for the case of a vertical engine with late ignition, the offset is 37.0 per cent, which is nearly the average of 20 per cent and 50 per cent, as might have been expected from the inertia constants.

11 Fig. 2 shows the decrease or increase in average side pressure or total loss of work from side pressure, in per cent of lost work with zero offset. While the use of a different indicator card with a later ignition might have made some difference, it obviously could not have changed the result so materially as in the case of maximum side pressure. Thus the curve marked four-cycle vertical may be fairly compared with the result in Table 9 of the paper, that the most favorable offset lies between 30 and 50 per cent.

12 The curves marked 450 and 1000 r.p.m. show the results of the few computations concerning these speeds not considered in Fig. 1.

13 Both Table 1 and Fig. 2 appear to indicate that more could be gained from offsetting with a two-cycle than with a four-cycle engine. But at present it is difficult to make general statements about this type of engine, and whether or not this advantage will be attained will depend upon the inertia constant and indicator card shown by these engines.

PRINCIPAL CONCLUSIONS

a An offset cylinder may be employed with least benefit on a high-speed four-stroke cycle vertical engine.

b It may be employed with most benefit on a slow-speed two-stroke cycle horizontal engine.

c The maximum advantageous offset is limited by the side pressure near the beginning of the first stroke.

JOHN H. NORRIS. I have been designing and building both two-stroke and four-stroke cylinder engines with offset cranks for a numsber of years. Our concern was so impressed with the advantage that in 1886 they bought the patent right to apply the offset stroke to gas engines. We are still building a few small sizes with offset crank, but there is no practical gain and as fast as we can re-design the engines we find we can get better economy and a more convenient engine, by placing the cylinder directly over the crank shaft. If you want to reverse the direction of rotation of an engine with an offset crank, you are in trouble at once. We have built an offset engine as large as four-cylinder, 14 by 22, and scrapped. I would like

to say, in connection with the large engine above mentioned, that it was built in 1896 and was one of a pair that were to run a suburban electric railroad in the West, on a suction gas producer. We built at that time a suction producer that was reasonably satisfactory. We have had a great deal of success, however, with our small single and double-cylinder engines with offset cranks, of which we have built a large number, though we are just putting on the market engines to replace them, in which we have placed the cylinder directly over the center. We used an offset of from 20 to 25 per cent of the stroke. With this offset the side strain is quite sufficient on the upstroke. These were all vertical engines.

THE AUTHOR would suggest that the fact that the cylinders were offset was not the real cause for scrapping the 4-cylinder 14 by 22 engine mentioned by Mr. Norris. Of course if stock engines were built, some to rotate in one direction and some in the other, or if it is desirable to build engines that may be reversed, offsetting may not practically be taken advantage of, as Mr. Norris points out. Furthermore, in spite of Mr. Norris' experience and his desire to eradicate offsetting from his product, this practice, in small vertical 4-cycle automobile engines, at least, seems to be increasing.

COMMENT ON CURRENT BOOKS

AUTOMATIC SCREW MACHINES AND THEIR TOOLS. By C. L. Goodrich and F. A. Stanley. *Hill Publishing Co., New York*, 1909. 8vo., cloth, vii + 255 p., illustrated. Price $2 net.

In the May number of The Journal a book by these authors on Accurate Tool Work was reviewed, and the present volume is of very much the same character, but dealing with another department of machine and tool work. Mr. Goodrich is department foreman with the Pratt and Whitney Co., Hartford, Conn., and Mr. Stanley is associate editor of the American Machinist. Both authors are expert tool-makers and designers and the matter which they have presented has been contributed mainly by them and to some extent by others, to the American Machinist, but revised, rewritten and added to for publication in book form. The volume is prepared in two parts, the first dealing with different types of machines and their special tools and the second with the design and construction of tools in detail, and their use.

Contents, by chapter headings: The Pratt & Whitney Automatic Screw Machine; Camming the Pratt & Whitney Automatic Screw Machine; The Brown & Sharpe Automatic Screw Machine; Laying out the Brown & Sharpe Screw Machine Cams; The Brown & Sharpe Automatic Screw Machine with Constant-Speed Drive; The Cleveland Automatic Turret Machine and its Cam Adjustments; The Gridley Automatic Turret Lathe; The Alfred Herbert Automatic Screw Machine; The New Spencer Double-Turret Automatic Screw Machine; The Cleveland Double Spindle Plain Automatic Machine; The Acme Multiple-Spindle Automatic Screw Machine; The Universal Multiple-Spindle Automatic Screw Machine; The Cleveland Automatic Machine with Magazine Attachment; The Alfred Herbert Magazine Automatic Screw Machine; The Potter & Johnston Automatic Chucking and Turning Machine; The Gridley Semi-Automatic Piston Ring Machine; The Prentice Multiple-Spindle Automatic Turret Machine; Points in Setting Up and Operating Automatic Screw Machines; Speeds and Feeds for Screw Machine Work; Spring Collets and Feed Chucks; Box Tools and Other External Cutting Appliances; Drills, Counterbores and Other Internal Cutting Tools; Screw Machine Taps and Dies; Forming Tools and Methods of Making Them; Nurling Tools and Their Applications; Why Chips Cling to Screw Machine Tools.

INTERNAL COMBUSTION ENGINES, Their Theory, Construction and Operation. Tables. Formulæ. 379 illustrations. By Rolla C. Carpenter, M.M.E., LL.D. and H. Diederichs, M.E. *D. Van Nostrand Company, New York*, 1909. Second edition revised. Cloth, 579 p. Price $5.

The theoretical principles of the internal combustion engine and their application to practical construction are treated but the proportioning and the strength of parts are not considered. The first five chapters are taken up with definitions and theory. Following this are three chapters treating respectively of solid, liquid and gaseous fuels. In the first the most important types of producers are illustrated and described. The chapter on liquid fuels deals largely with various vaporizing devices for crude oil, gasoline, kerosene and alcohol, the last-named in some detail. Valuable data will be found in the chapter on cost of installation and operation. One table compares the costs of 200-h.p. and 120-h.p. steam and gas plants. This second edition is a thorough revision of the first edition published a year ago.

Contents by chapter headings: Introduction, Definitions and Classifications, Indicated and Brake Horsepower; Thermodynamics of the Gas Engine; Theoretical Comparison of Various Types of Internal Combustion Engines; The Various Events of the Constant-Volume and Con-

stant-Pressure Cycle as Modified by Practical Conditions; The Temperature Entropy Diagram Applied to the Gas Engine; Combustion; Gas-Engine Fuels, The Solid Fuels, Gas Producers; The Gas Engine Fuels—Liquid Fuels; Carbureters and Vaporizers; Gas-Engine Fuels,—The Gas Fuels; The Fuel Mixture,—Explosibility, Pressure and Temperature; The History of the Gas Engine; Modern Types of Internal Combustion Engines; Gas Engine Auxiliaries—Ignition, Mufflers and Starting Apparatus; Regulation of Internal Combustion Engines; The Estimation of Power of Gas Engines; Methods of Testing Internal Combustion Engines; The Performance of Gas Engines and Gas Producers; Cost of Installation and of Operation.

THE MODERN ASPHALT PAVEMENT. By Clifford Richardson, M.Am.Soc.C.E., *John Wiley & Sons, New York*, 1908. Second edition, revised and enlarged. Cloth, 6 by 9; ix + 629 p.; 42 illustrations. Price $3.

The methods adopted by the author for the presentation of his material make this book of value alike to the engineer, the contractor, the road commissioner and the taxpayer. The descriptions of how asphalt pavement is made will be readily understood by the lay reader, while the substance of those chapters—on the chemistry of asphalt and the technology of the industry—intended for the engineer and expert may be had from the summary at the end of each chapter.

Contents by chapter headings: The Foundation or Base; The Intermediate Course; The Mineral Aggregate; Filler or Dust; The Nature of the Hydro-carbons which Constitute the Native Bitumens; Characterization and Classification of the Native Bitumens; Differentiation and Characterization of the Native Bitumens; Petroleums; The Solid Bitumens.

PRACTICAL TESTING OF GAS AND GAS METERS. By C. H. Stone, B.S., M.S. *John Wiley & Sons, New York*, 1909. Cloth, 6 by 9; x + 337 p.; 51 illustrations. Price $3.50.

The author's work as chief inspector of gas for the Public Service Commission, Second District, New York, has enabled him to obtain valuable data on the subject treated. As the title indicates, the book deals with methods of testing rather than manufacture and distribution. The chemical processes and calculations have been explained at length so that they may be clear to the gas-works manager and others with no chemical training. The author makes such comments and suggestions as will be helpful in the choice of apparatus or process. So far as we know, American literature on this subject is confined to the volume before us.

Contents by chapters:The Photometer and Accessories; Standards and Burners; Candle-Power Tests of Coal and Water Gas, Using Candles as Standards; Photometric Work with other Standards and Gases; Interpretation of Results and Legal Requirements; Carbonic Acid and Sulphuretted Hydrogen; Total Sulphur; Other Impurities; The Analysis of Gas; The Junker and Boys Calorimeters; Other Instruments and Methods; Consideration of Results; Specific Gravity and Pressure; The Cubic Foot and Meter Prover; The Method of Testing Meters.

STEAM POWER PLANT ENGINEERING. By G. F. Gebhardt. *John Wiley & Sons, New York*, 1908. Cloth, 6 by 9, xxix + 816 p.; 461 illustrations. Price $6.

The book contains the substance of a series of lectures at the Armour Institute of Technology. The author devotes fully a third of the book to matters intimately connected with the coal pile. In the chapter on fuels, the use of oil as a fuel receives attention. Somewhat incomplete figures are given of the cost of boilers, stokers and chimneys. An analysis is given of the strains in a steel-concrete chimney. Considerable space is devoted to the apparatus of the power plant, including steam engines, turbines and auxiliaries, cost figures of which are given. Perhaps the chapter of greatest interest to engineers will be that on the cost of power. A very complete bibliography supplements the numerous references in the text.

Contents by chapter headings: Elementary Steam Power Plants; Fuels and Combustion; Boilers; Smoke Preventions, Furnaces, Stokers; Superheated Steam, Superheaters; Coal and Ash-Handling Apparatus; Chimneys; Mechanical Draft; Steam Engines; Steam Turbines; Con-

densers; Feed-Water Purifiers and Heaters; Pumps; Separators, Traps and Drains; Piping and Pipe Fittings; Lubricants and Lubrication; Finance and Economics—Cost of Power; Testing and Measuring Instruments; Typical Specifications; A Typical Steam Turbine Station; A Typical Isolated Station. Appendices: General Bibliography—Power Plant Engineering and Design; A. S. M. E. Rules for Conducting Boiler Trials, Code of 1899; A. S. M. E. Rules for Conducting Steam Engine Tests; Steam Tables; Equivalent Values of Mechanical and Electrical Units; Miscellaneous Conversion Tables.

REFRIGERATING MACHINES; COMPRESSION; ABSORPTION. By Gardner T. Voorhees, S.B. *Ice and Cold Storage Publishing Company, Ltd., London*, 1909. Cloth, 5 by 7½; 70 p.; 10 diagrams, 4 plates. Price 4*s.* 6*d.*

The book is a reprint of a paper read before the First International Congress of the Refrigerating Industries at Paris in 1908. Mr. Voorhees attended this congress as a delegate of the United States government and was also appointed Honorary Vice-President of the A. S. M. E. to represent the Society on the occasion. He has included in the reprint a broader treatment of the subject than would ordinarily be found in a book of this size, as must necessarily be the case to come within the limits of a paper. He has included data from his own practice to enable engineers readily to solve problems of capacity and economy of a compression or an absorption system, or of their combinations. A number of tables and charts are included.

ELECTRIC MOTORS. THEIR INSTALLATION, CONTROL, OPERATION AND MAINTENANCE. By Norman G. Meade. *McGraw Publishing Company, New York*, 1908. Cloth, 5 by 7; v + 159 p.; 126 illustrations. Price $1.

The treatment is somewhat elementary, and mathematics have been eliminated. The illustrations and diagrams with the text serve to explain the different types of motors and their operation.

Contents by chapter headings: Theory of Electric Motors; Classification of Electric Motors; Control and Auxiliary Apparatus; Motor-Generator Sets; Installation of Electric Motors; Operating Hints; Repairs; Tables and Formulæ.

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R. WOLF, *Magdeburg, Germany*. A 500–650 h.p. patent reciprocating steam engine.

EMPLOYMENT BULLETIN

The Society has always considered it a special obligation and pleasant duty to be the medium of securing better positions for its members. The Secretary gives this his personal attention and is most anxious to receive requests both for positions and for men available. Notices are not repeated except upon special request. Copy for notices in this Bulletin should be received before the 15th of the month. The list of men available is made up of members of the Society, and these are on file, with the names of other good men not members of the Society who are capable of filling responsible positions. Information will be sent upon application.

POSITIONS AVAILABLE

073 Supervisor of labor for factory employing 500 men. Line machine tools. Location Ohio. Applicant must be specially qualified to handle all matters pertaining to the employment and efficiency of labor. Give full particulars regarding qualifications, stating experience in detail, especially during past five years; state age and education.

MEN AVAILABLE

277 Member, graduate M. I. T., nine years' experience in mechanical engineering instruction and practice, desires commercial engagement in or near Chicago, Strength of materials and testing a specialty.

278 Member, with fifteen years' experience, expert on gas engines, producers and furnaces, gasoline and oil engines, pumping machinery; both designer and superintendent. At present chief engineer of medium-sized shop. University graduate, best of references.

279 Junior, experienced in design of machines used in paper, rubber, food, chemical and allied industries; at present located in middle west, desires position in or near New York City.

280 Member, age 40, desires position as assistant superintendent or designer of special machinery. Has had experience with paper bag and envelope machinery, steam disinfecting and sterilizing apparatus. Nine years superintendent of large blower works, four years with largest manufacturers of power transmission apparatus.

281 Mechanical and civil engineer, technical graduate, energetic, aggressive and systematic, 16 years experience as designer, chief draftsman and engineer. Conversant with both mechanical and structural work. Desires responsible position with first-class firm in engineering or executive capacity.

282 Junior member, graduate mechanical engineer, at present draftsman for company manufacturing gasoline and electric autos and motor buggies; desires position with automobile concern which will yield experience in all branches of the business and offer opportunity for advancement. Experienced especially in automobile and electrical engineering. Can demonstrate and sell, design and superintend construction.

283 Electrical and mechanical engineer, Cornell graduate, age 32. Practical experience power plant and shop superintendence; electric and pneumatic power distribution; application of electricity in manufacturing plants; specification work and correspondence. Broad knowledge of general machinery; executive and business ability; highest endorsements.

284 Junior member, at present engaged with company operating blast furnaces, mines etc., familiar with manufacture of merchant pig iron, including Gayley Dry Blast, and all details entering into plant operation; can handle men and produce results.

285 Junior member would like position as instructor in mechanical engineering. Graduate M. I. T., taught in technical schools for over two years, and with experience in engineering practice.

286 Gas and power plant engineer open for an engagement (for two years). Salary $6000 per annum. Specialty, construction and reorganization.

287 Recent graduate Stevens Institute, desires to locate with concern engaged in manufacture of gas or steam engines, turbines, etc. Will start in foundry or shop.

288 Superintendent of blast furnaces and rolling mills of large Pennsylvania corporation, drafting, rolling-mill design and machinery sales. Experience covers twenty years.

289 Member, fifteen years' experience on engines, boilers, condensers, heaters, piping, and power plants. Has finished present construction work and desires engagement in New England with manufacturer or as representative on salary or commission. Age thirty-nine. First-class references.

290 Mechanical engineer, having large practical foundry and machine shop experience, as well as technical education, is open for temporary engagement to investigate and adjust complaints, superintend installations of machinery, or furnish reports on examination of machines, power and industrial plants.

291 Junior member, just returned from an extended trip of study and inspection of railroad and shop practice in Europe. Has had three years' practice in locomotive and gas engine constructions. Seeks foreign representation of some machine tool manufacturing company. Speaks German. Member of German Society of Engineers. Would consider a railroad position in operating or motive power department.

292 Massachusetts Institute of Technology course in mining and metallurgy, also mechanical engineering one year. Varied practical experience as draftsman, car designer, superintendent, manager, executive positions, in charge of buying, selling, works management and entire charge of business employing maximum of 700 men. Have patented about 50 inventions particularly relating to structural details of railroad cars.

293 Member, technical graduate, thirteen years' practical experience, drafting machine tools, cranes, etc., heavy-duty steam, gas and special heat-engine design, testing and operation, also operation of large A.C and D.C. power plants; desires position as chief engineer or designer for engine builder, or as master mechanic of large plant.

294 Sales engineer for special machinery or plant equipment. Broad executive and shop experience, accurate estimator. Competent to manage office or plants. Eastern states preferred.

295 Works manager or superintendent,thoroughly experienced in modern machine shop, foundry and manufacturing practice, office systems, etc. Successfully filled similar positions for fifteen years. Good organizer and can turn out high-grade work at low cost.

296 Mechanical engineer with six years' experience in industrial engineering, will be open for engagement about October 1. Desires position in or near New York City.

297 Graduate M. I. T., experience in shop, drafting and as mechanical engineer for large manufacturers of shoe machinery, including wide range of work on design of interchangeable machinery; experience in building extensions to factory, rearrangement of machinery, etc. Would like position of further responsibility either in engineering or manufacturing line. Some teaching experience.

298 Junior member, graduate in mechanical engineering, University of Illinois; competent to engage as factory manager, superintendent or chief engineer. Experience as instructor in charge of shops and mechanical engineering course.

299 Experienced superintendent modern shop and foundry. Qualified to take charge of production or full management. Technical and practical. Good organizer and executive.

CHANGES IN MEMBERSHIP

CHANGES OF ADDRESS

ACKERMAN, Wm. S. (1891; 1895; 1899), Cons. Engr., Colt Bldg., Paterson, N. J.

ADAMS, Walter H. (Junior, 1908), Prof. Mech. Engrg., Imperial Pei-Yang Univ., Tientsin, China.

ARNOLD, George (1904), Cleveland Frog and Crossing Co., and *for mail*, 7108 Hough Ave., Cleveland, O.

BARY, Mark (Associate, 1903), care of M. F. Levy, 621 Symes Bldg., Denver, Colo.

BEECHER, J. F. (Associate, 1908), Draftsman, Pa. Steel Co., and *for mail*, 203 State St., Harrisburg, Pa.

BIGELOW, Charles H. (1904), with Chas. T. Main, 45 Milk St., Boston, Mass.

BLACK, Edward S. (1903), 1407 Ry. Exchange Bldg., Chicago, Ill.

BLUMGARDT, Isaac E. (Associate, 1908), Hotel Florence, Tarrytown, N. Y.

BRADSHAW, Grant D. (Junior, 1904), Cons. Engr., 77 Jackson Blvd., Chicago, Ill.

BRENDLINGER, Wm. B. (Associate, 1907), Mgr., Ingersoll-Rand Co., 1226 Farmers' Bank Bldg., Pittsburg, Pa.

BROOKS, Paul Raymond (Junior, 1905), Union Bag and Paper Co., Sandy Hill, N. Y.

BROWN, Donald S. (Junior, 1908), Westinghouse Elec. and Mfg. Co., Pittsburg, Pa.

BULKLEY, J. Norman (1902), Cons. Elec. Engr., Genl. Mining and Finance Corp., and *for mail*, P. O. Box 1242, Johannesburg, Transvaal, South Africa.

BURGOON, Charles Eli (1907), Asst. Inspecting Engr., Isthmian Canal Com., and 18 Iowa Circle, Washington, D. C.

CARACRISTI, Virginius Z. (1904; 1907), 2205 Hanover Ave., Richmond, Va.

CARSTENS, Alexander Bismarck (1906), Gen. Mgr., Cia Minera del Sur. S. A., Sola de Vega, and Mina San Pablo, Teojomulco, and *for mail*, Apartado Postal 74, Oaxaca, Mexico.

CHACE, William W. (1908), Mech. Engr., Cleveland Twist Drill Co., and *for mail*, 9716 Logan Court, Cleveland, O.

CHAMBERLAIN, George E. (1907), La Grange, Ill.

CHRISTIE, Alexander G. (Associate, 1907), Asst. Prof. Steam Engrg., Univ. of Wisconsin, and 1713 Monroe St., Madison, Wis.

CHURCH, Elihu C. (Junior, 1908), Siasconset, Nantucket Is., Mass.

CLEMENS, A. B. (1904), Prin., Sch. Mech. Engrg., Internatl. Corr. Schools, and 1003 Quincy Ave., Scranton, Pa.

COLVIN, Fred Herbert (1895; 1899), Assoc. Editor, Am. Machinist, 505 Pearl St., New York, N. Y., and *for mail*, 56 N. Maple Ave., East Orange, N. J.

CROSBY, William Wyman (1900; 1902), with F. W. Dean, Mill Engr. and Arch., Rm. 1112, 53 State St., Boston, and 41 Arlington Rd., Woburn, Mass.

DAVIS, Charles Ethen (1896), Genl. Mgr., Warner Gear Co., Muncie, Ind.

DECKER, Edward P. (1906), Engr., Kemiweld Can Co., and 79 Pingree Ave., Detroit, Mich.

DODDS, William B. (Junior, 1907), Cons. Light, Power and Ice Co., Joplin, Mo.

DOHERTY, Henry L. (1903), Pres., Denver Gas and Elec. Co., 60 Wall St., New York, N. Y.

DONNELLY, William T. (1903), Cons. Engr., 135 Broadway, New York, N. Y.

DOUD, Arthur T. (Junior, 1907), Mech. Engr., Hale-Kilburn Metal Co., Glenwood Ave. and 19th St., Philadelphia, and 6365 Sherman St., Germantown, Philadelphia, Pa.

DUNN, Charles (1897), Hanover, N. H.

ELDRED, Byron E. (1899; 1903), Pres., Commercial Research Co., 149 Broadway, New York, and Box 4, Tuckahoe, also Bronxville, N. Y.

ELVIN, Albert Geo. (1907), Asst. to V. P., Franklin Ry. Supply Co., 30 Church St., New York, N. Y.

FLETEMEYER, Louis H. (Junior, 1906), 1425 Salem St., Lafayette, Ind.

GAY, Harry (Associate, 1907), A. L. Drum & Co., 624 Am. Trust Bldg, and *for mail*, 7111 Princeton Ave., Chicago, Ill.

GERRISH, William H. (1901), Supt. Soft Fibre Dept., Columbian Rope Co., and *for mail*, Osborne House, Auburn, N. Y.

HAGERTY, Walter W. (Junior, 1905), P. O., Middletown, Orange Co., N. Y.

HEARNE, Robert J. (1907), Secy. and Treas., Durbrow & Hearne Mfg. Co., 12 Wooster St., and 46 E. 82d St., New York, N. Y.

HELVEY, Geo. Stanley (Junior, 1904), Mgr. of Shops, Imperial Motor Car Co., and *for mail*, 303 N. 2d St., Hamilton, O.

HENRY, Geo. J., Jr. (1901), Ch. Engr., Pelton Water Wheel Co., Rm. 1099 Monadnock Bldg., and 3999 Clay St., San Francisco, Cal.

HERRICK, James A. (1880), Rm. 1603, 2 Rector St., New York, N. Y.

HERRON, James H. (1897; 1905), Mgr., Motch & Merryweather Mchy. Co. 1124 Farmers' Bank Bldg., Pittsburg, and Elmhurst Inn, Sewickley, Pa.

HIGGINS, Albert W. (Associate, 1906), Engr., Hill-Hupfel Engrg. Co., 30 Church St., New York, N. Y.

HILL, Thomas (1900), Cons. Engr., 2102 Jersey St., and 2102 Lawndale, Quincy, Ill.

HOLLMANN, Frederick W. (Junior, 1907), Asst. Mech. Engr., Carborundum Co., Niagara Falls, N. Y.

IRELAND, Mark L. (Junior, 1902), 1st. Lieut., Ordnance Dept., U. S. A., Watertown Arsenal, Watertown, Mass.

JAKOBSSON, Herman G. (1907), Ordnance Designer, Rm. 135, Bureau of Ordnance, War Dept., and 3 Stanley Apts., Harvard St., Washington, D. C.

JETT, Carter C. (Junior, 1902), 638 Oak Hill Ave., Youngstown, O.

JOHNSON, J. E., Jr. (1896; 1901), Genl. Supt., Thomas Div., Republic Iron and Steel Co., Thomas, Ala.

JONES, Jarrard E. (Junior, 1903), U. S. S. South Carolina, Cramp's Shipyard, Philadelphia, Pa.

JURGENSEN, Jess Christian (Associate, 1905), 29 W. 97th St., New York, N. Y.

KEELY, Royal R. (1901; 1907), Prof. Elec. Engrg., Nova Scotia Tech. College, Halifax, N. S.

KEMBLE, Parker H. (1908), 62 Montague St., Brooklyn, N. Y.

KENT, Wm. (1880), Manager, 1885–1888; V. P., 1888–1890; 326 Washington St., Sandusky, O.

KNOX, S. L. Griswold (1892; 1901), V. P. and Ch. Engr., Bucyrus Co., South Milwaukee, Wis.

LARNER, Chester W. (Associate, 1907), Hyd. Engr., Wellman-Seaver-Morgan Co., and *for mail*, 6915 Euclid Ave., Cleveland, O.

LATTA, Nisbet (Junior, 1902), Mill and Gas Eng. Dept., Allis-Chalmers Co., West Allis, Wis.

LEE, Ralph A. (Junior, 1909), Asst. Bldg. Supt., with Walter Kidde, 140 Cedar St., New York, and *for mail*, 414 78th St., Brooklyn, N. Y.

LEE, Wm. F. (1907), 1111 E. Main St., Richmond, Va.

LELAND, Frederic H. (1907), Production Engr., 122 Elmwood Ave., Narberth, Pa.

LEWIS, Wm. Y. (1902), 20, Pennard Mansions, Shepherds Bush, London, W., England.

LILLIBRIDGE, Ray D. (Associate, 1907), Suite 20, 100 Broadway, and P. O. Box 824, New York, N. Y.

LOCKWOOD, Burns D. (1907), Asst. Ch. Engr., Pressed Steel Car Co., McKees Rock, and *for mail*, 444 Dawson Ave., Bellevue, Pa.

LYMAN, Elihu Root (Junior, 1904), Mech. Engr., U. S. Heater Co., and *for mail*, 934 Lafayette Blvd., Detroit, Mich.

MacINTIRE, Horace Jas. (Junior, 1907), Instr. Engrg. Dept., Harvard Univ., Cambridge, Mass.

MacLAUCHLAN, James H. (1907), Engr., Am. Cement Engrg. Co., Yorktown, Va.

MacPHERSON, James Day (1899), Ch. Engr., Adolphus Busch, Purchaser of Am. Diesel Eng. Co., and *for mail*, 3513 Halliday Ave., St. Louis, Mo.

MORGAN, Lewis Henry (1904; 1906), Nuthurst, Nantwich, Cheshire, England.

MORITZ, Albert (1905), Comdr. U. S. Navy, Ret., 723 E. 18th St., Brooklyn, N. Y.

NAYLOR, Chas. Wm. (1905), Ch. Engr., Marshall Field & Co., 83 State St., and *for mail*, 4429 Congress St., Chicago, Ill.

NEWCOMB, Chas. L., Jr. (Associate, 1908), Deane Steam Pump Co., and *for mail*, 57 Fairfield Ave., Holyoke, Mass.

NICKERSON, Ralph R. (Junior, 1907), 65 Otis St., East Cambridge, Mass.

NICKLIN, Ernest W. (1900; Associate, 1907), Diamond Specialty Co., and 421 Cadillac Ave., Detroit, Mich.

OSBOURN, Millard P. (Junior, 1901), Osbourn & Robinson, 320 Penna. Bldg., Philadelphia, Pa.

PEARSON, Walter Ambrose (1907), Asst. Genl. Mgr., Rio de Janeiro Tramway Lgt. and Power Co., Avenida Central 76, Rio de Janeiro, Brazil, South America.

PENNINGTON, James H. (1902), Mech. Engr., Supt. of Cons. and Power, Am. Smelting and Refining Co., Perth Amboy Plant, Maurer, and *for mail*, 1 Lewis St., Perth Amboy, N. J.

PINGER, Geo. C. (Junior, 1907), Driggs-Seabury Ordnance Corps, Sharon, Pa.

RENNER, Roland B. (Junior, 1906), 226 Buttles Ave., Columbus, O.

RUPP, Manning E. (Junior, 1909), Gatun, Canal Zone, C. A.

SARENGAPANI, T. S. (Junior, 1903), Head Draftsman P. W. D., Guntur Dist., Madras, India.

SAR VANT, Wilbur N. (Junior, 1907), 109 Halsey St., Newark, N. J.

SERGEANT, Chas. H. (1895), 625 W. 138th St., New York, N. Y.

SHENBERGER, Geo. H. (Junior, 1909), Cast Thread Fitting and Fdy. Co., Seneca Falls, N. Y.

SINCLAIR, Angus (1883), Editor and Publisher, 114 Liberty St., New York, N. Y., and *for mail*, Millburn, N. J.

SPRADO, Ralph (1900), 1604 San Antonio Ave., Alameda, Cal.

STOUGHTON, Edwin R. (Associate, 1907), The Charlevoix, Apt. 1107, Detroit, Mich.

STRATTON, H. F. (Junior, 1904), Sales Mgr., Elec. Controller & Mfg. Co., and *for mail*, Suite 7, 1523 E. Boulevard, St. Louis, Mo.

STROM, Carl A. (1905), Mgr., Rogers Wks., Am. Loco. Co., Paterson, N. J.

STURGESS, John (1901), Hyd. Engr., 9 Arch St., Akron, O.

SULLIVAN, Lucien N. (1903), Translator War Dept., Army War College, and *for mail*, 2207 1st St., N. W., Washington, D. C.

TADDIKEN, J. F., Jr. (Junior, 1907), Am. Beet Sugar Co., Chino, Cal.

TAKEO, Toshisuke (1908), Cons. Engr., Yoshinotani Coal Mining Co., Ltd., Genl. Mgr., Karatsu Iron Wks., Karatsu, Hizen, Japan.

TOWN, Frederic E. (1905), Otis Elevator Co., Keenan Bldg., Pittsburg, Pa.

VALENTINE, Warren P. (Junior, 1904), Gen. Delivery, Holyoke, Mass.

VOSE, Fred Hale (Junior, 1906), Mech. Engrg. Dept., Case Sch. of Applied Science, and *for mail*, 10521 Lee Ave., Cleveland, O.

WARG, Robert (1903), Natl. Blower Wks., 17th St. and St. Paul Ave., and 319 Farwell Ave., Milwaukee, Wis.

WARREN, Ambrose Gilmore (1907), Mech. Engr., J. W. Paxson Co., 1607 Mt. Vernon St., Philadelphia, Pa.

WATSON, George L. (Junior, 1905), P. O. Box 448, Dover, Del.

WOLDENBERG, Izydor (Junior, 1903), Ingersoll-Rand Co., m.b.H. Ferencz Jozsef Rakpart 1F, Budapest, Austria-Hungary.

WRIGHT, Reginald A. (1907), Ch. Draftsman, Phila. Coal and Iron Co., and *for mail*, 7 S. 6th St., Pottsville, Pa.

YARYCAN, Edward B. (Junior, 1903), Supt., Yaryan Naval Stores Co., Gulfport, Miss.

YATES, Richard Llewellyn (Junior, 1907), Ch. Draftsman, Waterwheel and Turbine Pump Dept., Platt Iron Wks. Co., and *for mail*, 108 Oxford Ave., Dayton, O.

DEATHS

GILDERSLEEVE, David H.

LANDSING, Jang.

REED, Wm. T.

NEW MEMBERS

AHLQUIST, Harold (1909), Designing Engr., Solvay Process Co., Syracuse, N. Y.

BAENDER, Fred Geo. (Junior, 1909), Mech. Engr., P. O. Box 735, Spokane, Wash.

BLAISDELL, Benjamin H. (1909), Ch. Engr. and Supt., Manilla Elec. R. R. and Light Co., P. O. Box 451, Manila, P. I.

BLANCHARD, Arthur S. (Associate, 1909), Asst. to Mgr., Atha Steel Casting Co., Newark, and *for mail*, 25 Hampton Terrace, East Orange, N. J.

BLEYER, Chas. F. (Junior, 1909), Erecting Engr., Allis-Chalmers Co., Elyria, and *for mail*, 836 Cass St., Milwaukee, Wis.

BURT, Clayton R. (1909), Genl. Supt., Barber-Colman Co., 523 Fisher Ave., Rockford, Ill.

CASTLE, Samuel Northrup (Associate, 1909), Genl. Elec. Co., 30 Church St., New York, N. Y.

CLANCY, Geo. W. A. (Associate, 1909), Shop Insp., New Haven System, Readville, Mass.

FULLER, Ray W. (Associate, 1909), Supt., Scranton Stove Wks., Scranton, Pa.

GARVIN, George K. (1909), Pres., Garvin Mch. Co., Spring and Varick Sts., New York, N. Y.

JENKS, Glen F. (Junior, 1909), Capt. Ordnance Dept., U. S. A., Frankford Arsenal, Philadelphia, Pa.

MAYALL, E. L. (1909), Genl. Supt., Rock Island Plow Co., and 1003 2d Ave., Rock Island, Ill.

MORAT, Jacques (1909), Spuyten Duyvil, N. Y.

NICHOLL, John Seymour (Junior, 1909), Acting Mgr., F. W. Horne, Importer Am. Mchy., 70-C, Yokohama, Japan.

OTTO, Henry S. (Junior, 1909), Messrs. Munroe & Co., rue Scribe 7, Paris, France.

PINNER, Seymour Wm. (Junior, 1909), Instr., Univ. of Michigan, and *for mail*, 608 E. Madison St., Ann Arbor, Mich.

RUPP, Manning E. (Junior, 1909), Computer-Estimator, Isthmian Canal Com., Culebra, Canal Zone, C. A.

SHAW, James C. (1909), Engr. and Tech. Adviser, Kawasaki Dockyard Co., Ltd., Kobe, Japan.

SHIELDS, George Rex (Associate, 1909), Engr., Improved Equipment Co., 60 Wall St., New York, N. Y.

STACKS, H. Roy (1909), Supt., Geo. V. Cresson Co., 18th and Allegheny Ave., Philadelphia, Pa.

STEWART, Chas. E. (Junior, 1908), Asst. Prof. of Mechanic Arts, Tufts College, Tufts College P. O., Mass.

TOBIN, Robert P. (1909), Ch. Tech. Dept., Vacuum Oil Co., 49 Federal St., Boston, Mass.

WATERS, William L. (1909), Designing Engr., Westinghouse Elec. and Mfg. Co., Pittsburg, Pa.

WELLS, Robert G. (1909), Genl. Mgr., Tata Iron and Steel Co. Ltd., Sakchi, via Kalimati, B. N. Ry., India.

WICKS, H. B. Priestley (Junior, 1908), Genl. Mgr. Elec. Dept., Scott Bros. Ltd., and *for mail*, 21 Graffon St., Linwood, Christchurch, New Zealand.

PROMOTIONS

BIBBINS, James Rowland (1904; 1909), Engr. Appraisal Bureau, Public Service Com., 1912 Tribune Bldg., New York, N. Y.

COOKE, Morris Llewellyn (1903; 1909), Cons. Engr., 401 W. Walnut Lane, Germantown, Philadelphia, Pa.

DAVOUD, Vahram Y. (1906; Associate, 1909), Elec. Engr., Telluride Power Co., Provo, Utah, and Univ. Club, Buffalo, N. Y.

GROVER, Marcus A. (1903; 1909), Cons. Engr., Box 590, Birmingham, Ala.

GAS POWER SECTION

CHANGES OF ADDRESS

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LATTA, Nisbet (1908), Mill and Gas Engrg. Dept., Allis-Chalmers Co., West Allis, Wis.

MACPHERSON, James Day (1908), Ch. Engr., Adolphus Busch, Purchaser of Am. Diesel Eng. Co., and *for mail* 3513 Halliday Ave., St. Louis, Mo.

MONAHAN, Louis J. (Affiliate, 1908), V. P., Supt. and Mech. Engr., Termaat & Monahan Co., and 57 Waugoo St., Oshkosh, Wis.

SAGE, Darrow (Affiliate, 1908), Hudson & Manhattan R. R. P.H., 1st and Washington Sts., Jersey City, N. J.

VOSE, Fred Hale (1909), Mech. Engrg. Dept., Case Sch. of Applied Science, and *for mail*, 10521 Lee Ave., Cleveland, O.

NEW MEMBERS

ADAMS, Edward T., (1909), Mgr. and Ch. Engr., Gas and Mill Eng. Dept., Allis-Chalmers Co., Milwaukee, Wis.

ANDERSON, Harry Warfield (1909), Genl. Mgr., White Star Auto Co., and *for mail*, No. 6, Lenox Apts., 31 Porter Pl., Atlanta, Ga.

BAENDER, Fred Geo. (1909), Mech. Engr., P. O. Box 735, Spokane, Wash.

BREW, William P. (Affiliate, 1909), Commercial Engr., Westinghouse Mch. Co., 165 Broadway, New York, N. Y.

CHASE, Leon Wilson (1909), Prof. Farm Mechanics, Univ. of Neb., and 1245 N. 33d St., Lincoln, Neb.

COLLETT, S. D. (1909), V. P. and Eastern Mgr., Elev. Supply and Repair Co., 114–116 Liberty St., New York, N. Y.

CUMMINGS, Wm. Warren (1909), Asst. Mgr., Commercial Dept., Boston Cons. Gas. Co , 24 West St., Boston, and *for mail*, 15 Winter St., Woburn, Mass.

FISH, Walter C. (Affiliate, 1909), Mgr. Lynn Wks., Gen. Elec. Co., and 21 Wave St., Lynn, Mass.

FLEMING, Wills Maine (1909), Engr. in charge Drawing Room, Deane Branch, Internatl. Steam Pump Co., and *for mail*, 370 Cabot St., Holyoke, Mass.

GROVER, Marcus A. (1909, Cons. Engr., Box 590, Birmingham, Ala.

HALE, Frederick J. (1909), Eng. Dept., Westinghouse Mch. Co., East Pittsburg, and *for mail*, 1133 Franklin Ave., Wilkinsburg, Pa.

HAMILTON, Clinton A. (1909), Genl. Mgr. and V.P., Wisconsin Eng. Co., and *for mail*, 1033 Lake Ave., Racine, Wis.

HERRICK, James A. (1909), Rm. 1603, 2 Rector St., New York, N. Y.

HOWELLS, Charles (1909), Supt., Watson-Stillman Co., 50 Church St., and *for mail*, 600 W. 142d St., New York, N. Y.

LATHAM, Harry M. (1909), Mech. Engr., Am. Steel & Wire Co., and *for mail*, 94 Grove St., Worcester, Mass.

LEE, Francis V. T. (1909), Asst. Genl. Mgr., Pacific Gas & Elec. Co., 445 Sutter St., San Francisco, and *for mail*, 59th St., E. of College Ave., Oakland, Cal.

MICHEL, Arthur Eugene (1909), Adv. Engr., 1572 Hudson Terminal Bldg., New York, N. Y.

MOORE, Allen H. (1909), Genl. Elec. Co., and *for mail*, 352 Western Ave., Albany, N. Y.

OULD, John Geo. (1909), Supt. and Ch. Engr., Polhemus Memorial Clinic., Brooklyn, N. Y.

PATITZ, J. F. Max (1909), Cons. Mech. Engr., Allis-Chalmers Co., and *for mail*, 473 38th St., Milwaukee, Wis.

RICE, George W. (1909), Engrg. Dept., Olds Gas Power Co., and *for mail*, 513 Sparrow Ave., Lansing, Mich.

ROTH, Charles (Affiliate, 1909), Mech. Engr., Liquid Carbonic Co., Wells and Mich. Sts., Chicago, Ill.

SAMPSON, Chas. C. (Affiliate, 1909), Inspr. Gas Power Mchy., Ill. Steel Co., South Chicago, and *for mail*, 1475 E. 73d St., Chicago, Ill.

SCOFIELD, Edward H. (Affiliate, 1909), Engr. and Power Equip., Twin City Rapid Transit Co., and 3313 Lyndale Ave., S., Minneapolis, Minn.

SIMONS, Joseph Aiken (Affiliate, 1909), Mech. Engr., E. I. du Pont de Nemours Powder Co., 607 du Pont Bldg., Wilmington, Del.

SPRADO, Carl G. (1909), Engr. and Ch. Draftsman, Gas and Mill Eng. Dept., Allis-Chalmers Co., and *for mail*, 752 29th St., Milwaukee, Wis.

TROUT, Geo. Henry (Affiliate, 1909), Sales Mgr., Smith Gas Power Co., Lexington, O.

VAIL, Jesse A. (1909), Genl. Mgr., Fairbanks, Morse Mfg. Co., Beloit, Wis.

VINCENT, Jay Carter (Affiliate, 1909), Asst. Engr. Power and Equip., Twin City Rapid Transit Co., Minneapolis, Minn.

DEATHS

GILDERSLEEVE, David H.

STUDENT SECTION

CHANGES OF ADDRESS

BECKMANN, Harry T. (Student, 1909), 2475 Collingwood Ave., Toledo, O.
BROOKMAN, H. D. (Student, 1909), 511 Riverside Ave., Wellsville, O.
COYLE, J. F. (Student, 1909), 1311 Fones Ave., St. Joseph, Mich.
FINCH, Ellis J. (Student, 1909), 260 W. 136th St., New York.
FREYGANG, Gustav G. (Student, 1909), Engrg. Dept., Natl. Elec. Lamp Asso., 4411 Hough Ave., Cleveland, O.
HANDY, Robert G. (Student, 1909), 625 University Ave., Ithaca, N. Y.
HOLLENBERGER, Theo. J. (Student, 1909), 2564 N. Paulina St., Chicago, Ill.
LAWRENCE, John H. (Student, 1909), 13 Grove Pl., Albany, N. Y.
LOCKARD, J. P. (Student, 1909), 26 Church St., Plymouth, Pa.
LUSE, F. D. (Student, 1909), 973 E. 9th St., N., Portland, Ore.
MUDD, John Posey (Student, 1909), 415 Clinton Ave., Rochester, N. Y.
PAGE, W. K. (Student, 1909), Olean, N. Y.
PARCE, Donald H. (Student, 1909), 2614 Waring St., Berkeley, Cal.
RICHARDSON, L., Jr. (Student, 1909), 1018 Lexington Ave., Altoona, Pa.
ROSCOE, A. P. (Student, 1909), Oakley Ave., Ozone Park, L. I., N. Y.
SAWERS, Edward H. (Student, 1909), 548 Lake Ave., Rochester, N. Y.
SCHICKEL, Norbert H. (Student, 1909), 52 E. 83d St., New York.
STEWART, Arthur L. (Student, 1909), 42 Prince St., Rochester, N. Y.
SYMES, W. H. (Student, 1909), 326 W. Church St., Marion, O.
WILSON, Joseph S. (Student, 1909), 404 S. Clayton St., Wilmington, Del.

NEW MEMBERS

KELLOG, E. N. (Student, 1909), Vineland, N. J.
VAIL, A. G. (Student, 1909), Milton-on-the-Hudson, N. Y.
VAN ETTEN, Percy H. (Student, 1909), 465 Hawthorne Ave., Palo Alto, Cal.
ZIMMERMAN, Peter C. (Student, 1909), State Agr. College of Ore., Corvallis, Ore.

COMING MEETINGS

SEPTEMBER AND OCTOBER

Secretaries or members of societies whose meetings are of interest to engineers are invited to send in their notices for publication in this department. Such notices should be in the editor's hands by the 15th of the month preceding the meeting.

AMERICAN GAS INSTITUTE

October 20–22, Hotel Pontchartrain, Detroit, Mich. Secy., A. B. Beadle, 25 West 39th St., N. Y.

AMERICAN INSTITUTE OF MINING ENGINEERS

September 27–29, Spokane, Wash. Papers: Dust Explosions; Modern Progress in Mining and Metallurgy; Ventilating System of the Comstock Mines; Professional Ethics; Ruble Hydraulic Elevator; etc. Acting Secy., Joseph Struthers.

AMERICAN MINING CONGRESS

September 27–October 2, Goldfield, Neb. Secy., J. F. Callbreath, Jr., Denver, Colo.

AMERICAN PUBLIC HEALTH ASSOCIATION

October, Richmond, Va. Secy., C. O. Probst, Columbus, O.

AMERICAN RAILWAY BRIDGE AND BUILDING ASSOCIATION

October 19–21, annual convention, Jacksonville, Fla. Secy., S. F. Patterson, Boston & Maine Ry., Concord, N. H.

THE AMERICAN SOCIETY OF MECHANICAL ENGINEERS

October 12, monthly meeting, 29 W. 39th St., New York. September 11, St. Louis, Mo. Secy., C. W. Rice, 29 W. 39th St., New York.

AMERICAN STREET AND INTERURBAN RAILWAY ASSOCIATION

October 4–8, annual convention, Denver, Colo. Secy., Bernard V. Swenson, 29 W. 39th St., New York.

ASSOCIATION OF CAR LIGHTING ENGINEERS

October 5–8, annual convention, Chicago, Ill. Secy., G. B. Colegrove, 6250 Drexel Ave., Chicago, Ill.

CANADIAN MANUFACTURERS ASSOCIATION

September 14–16, annual meeting, Hamilton, Can.

CENTRAL ASSOCIATION OF RAILROAD OFFICERS

September 22–24, annual meeting, Cincinnati, O. Genl. Secy., O. G. Felter, Carew Bldg.

CENTRAL RAILWAY AND ENGINEERING CLUB OF CANADA

September 21, October 19, Prince George Hotel, Toronto, Ont. Papers: Filling and Washing out of Boilers, J. V. Jackson; Care and Maintenance of Elevators, J. Shales. Secy., C. L. Worth, Union Sta., Toronto.

COLORADO ELECTRIC LIGHT, POWER AND RAILWAY ASSOCIATION

October 7–9, annual convention, Denver, Colo. Papers: The Commercial

Field; Meters and Testing; Lamps and Illumination. Secy., J. C. Lawler, P. O. Box 938, Colorado Springs.

INTERNATIONAL ASSOCIATION OF MUNICIPAL ELECTRICIANS
September 14–16, annual convention, Atlantic City, N. J. Secy., Frank P. Foster, Corning, N. Y.

KANSAS GAS, WATER, ELECTRIC LIGHT AND STREET RAILWAY ASSO.
September 23–24, Wichita. Secy., J. D. Nicholson, Newton.

NATIONAL ASSOCIATION OF STATIONARY ENGINEERS
September 13–18, Columbus, O. Secy., F. W. Raven, 325 Dearborn St., Chicago, Ill.

NEW ENGLAND WATER WORKS ASSOCIATION
September 8–10, annual convention, New York. Secy., Willard Kent, 715 Tremont Temple, Boston, Mass.

NOVA SCOTIA SOCIETY OF ENGINEERS
September 9–10, annual convention, New Glasgow. Secy., S. Fenn, Halifax.

PACIFIC COAST GAS ASSOCIATION
September 21–23, San Francisco, Cal. Secy., John A. Britton, 925 Franklin St.

PENNSYLVANIA ELECTRIC ASSOCIATION
September 8–10, annual convention, Eagles Mere, Pa. Pres., L. H. Conklin, Scranton.

RAILWAY SIGNAL ASSOCIATION
October 12–14, Louisville, Ky. Secy., C. C. Rosenberg, 712 North Linden St., Bethlehem, Pa.

ROADMASTERS AND MAINTENANCE OF WAY ASSOCIATION
September 14–17, annual convention, Washington, D. C. Paper: Treated Ties, J. M. Meade. Secy., W. E. Emery, P. & P.U. Ry., Peoria, Ill.

ROYAL ARCHITECTURAL INSTITUTE OF CANADA
October 5–7, Toronto. Secy., Alcide Chausse, R.S.A., P. O. Box 259, Montreal, Que.

TEXAS RAILWAY CLUB
September 20, semi-annual meeting. Secy., T. H. Osborne, Pine Bluff, Ark.

TRAVELING ENGINEERS' ASSOCIATION
September 7–10, annual convention, Albany Hotel, Denver, Colo. Secy., W. O. Thompson, N. Y. C. Car Shops, East Buffalo, N. Y.

WESTERN ASSOCIATION OF ELECTRICAL INSPECTORS
October 26–28, annual meeting, Detroit, Mich.

MEETINGS TO BE HELD IN ENGINEERING SOCIETIES BUILDING

Date	Society	Secretary	Time
September			
1	Wireless Institute	S. L. Williams	7.30
2	Blue Room Engineering Society	W. D. Sprague	8.00
3	Explorers' Club	H. C. Walsh	8.30
4	Amer. Soc. Hungarian Engrs. and Archts.	Z. deNemeth	8.30
7	N. Y. Society Accountants and Bkprs.	T. L. Woolhouse	8.00
8	Optometrical Society of City of N. Y.	J. H. Drakeford	8.00
9	Illuminating Engineering Society	P. S. Millar	8.00
10	American Institute Electrical Engrs.	R. W. Pope	8.00

14	N. Y. Soc. Accountants and Bkprs	T. L. Woolhouse	8.00
17	New York Railroad Club	H. D. Vought	8.15
21	N. Y. Soc. Accountants and Bkprs	T. L. Woolhouse	8.00
22	Municipal Engineers of City of N. Y.	C. D. Pollock	8.15
27–29	Illuminating Engineering Society	P. S. Millar	All day
28	N. Y. Society Accountants and Bkprs	T. L. Woolhouse	8.00
28–30	Amer. Electro-Therapeutic Asso.	A. C. Geyser	All day
October			
1	Explorers' Club	H. C. Walsh	8.30
2	Amer. Soc. Hungarian Engrs. & Archts.	Z. deNemeth	8.30
5	N. Y. Soc. Accountants and Bkprs	T. L. Woolhouse	8.00
6	Wireless Institute	S. L. Williams	7.30
7	Blue Room Engineering Society	W. D. Sprague	8.00
8	American Institute Electrical Engrs.	R. W. Pope	8.00
12	N. Y. Soc. Accountants and Bkprs	T. L. Woolhouse	8.00
12	The American Society of Mech. Engrs.	C. W. Rice	8.00
13	Optometrical Society of City of N. Y.	J. H. Drakeford	8.00
14	Illuminating Engineering Society	P. S. Millar	8.00
15	New York Railroad Club	H. D. Vought	8.15
19	New York Telephone Society	T. H. Lawrence	8.00
19	N. Y. Society Accountants and Bkprs	T. L. Woolhouse	8.00
26	N. Y. Society Accountants and Bkprs	T. L. Woolhounse	8.00
27	Municipal Engineers of City of New York	C. D. Pollock	8.15

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Terms expire at Annual Meeting of 1909

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1909

On a Standard Tonnage Basis for Refrigeration

D. S. Jacobus — G. T. Voorhees
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E. F. Miller

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1909

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CHAS. WALLACE HUNT (1)
F. R. HUTTON (2)
FRED J. MILLER (3)

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J. W. LIEB, JR., CHAIRMAN OF THE LIBRARY COMMITTEE OF THE AM. SOC. M. E.

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NOTE.—Numbers in parentheses indicate length of term in years that the member has yet to serve.

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1909

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